

AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

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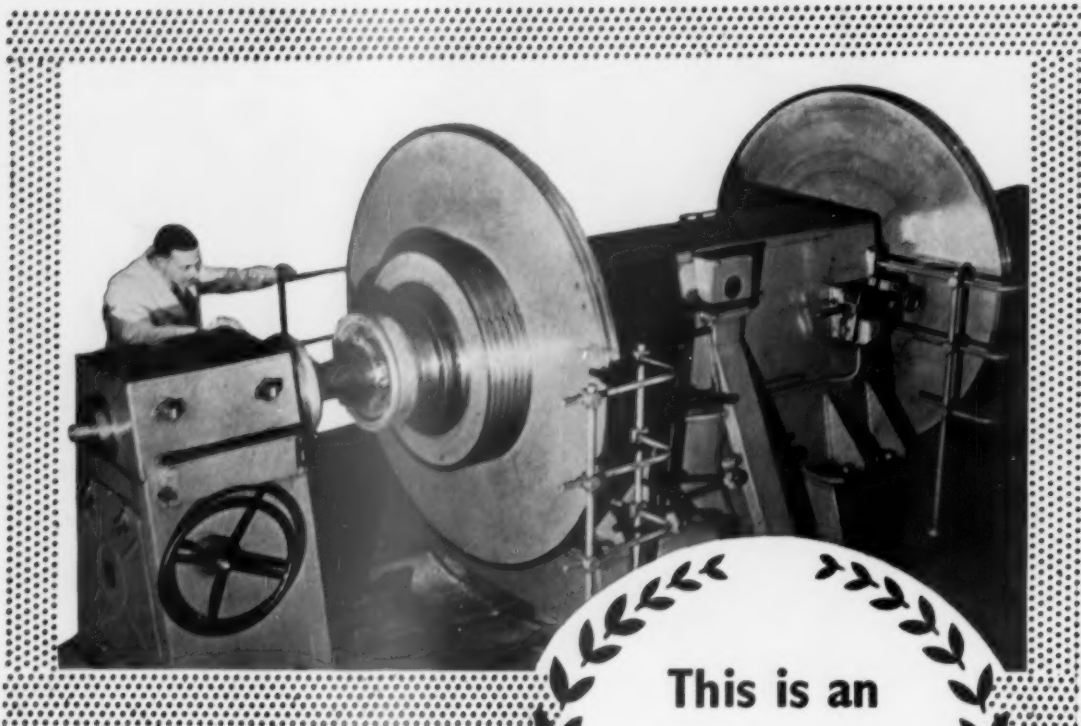


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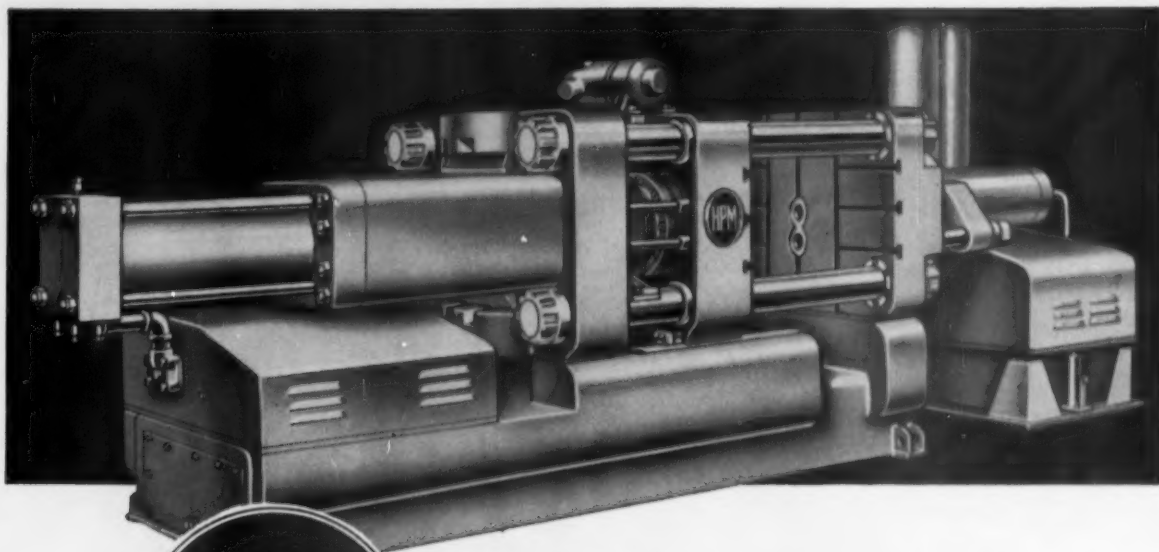
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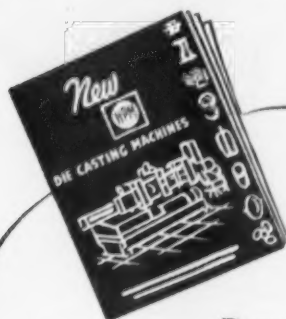
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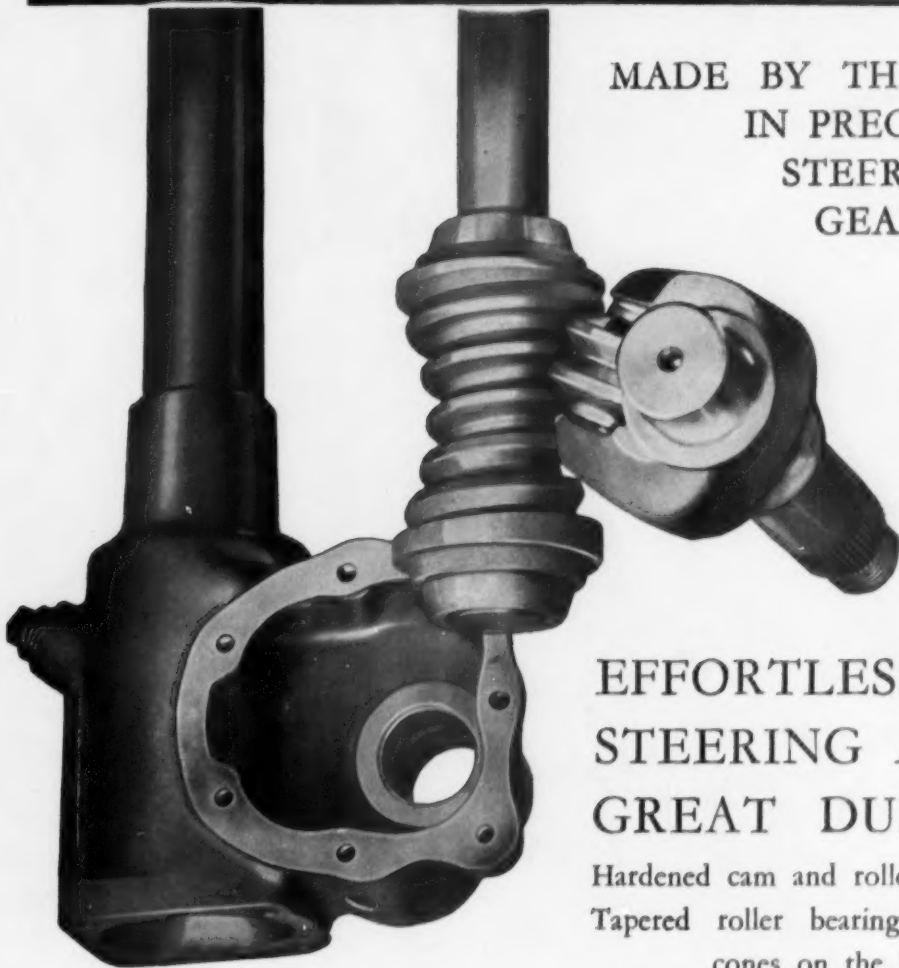
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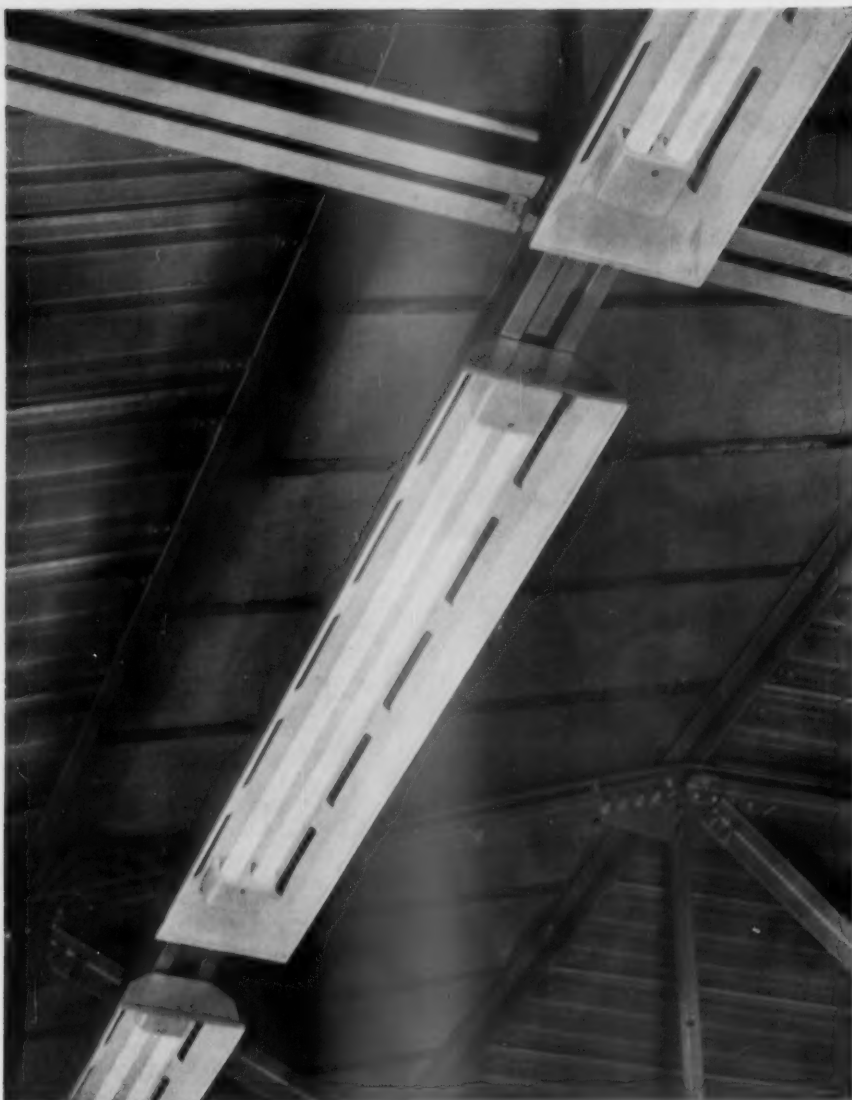
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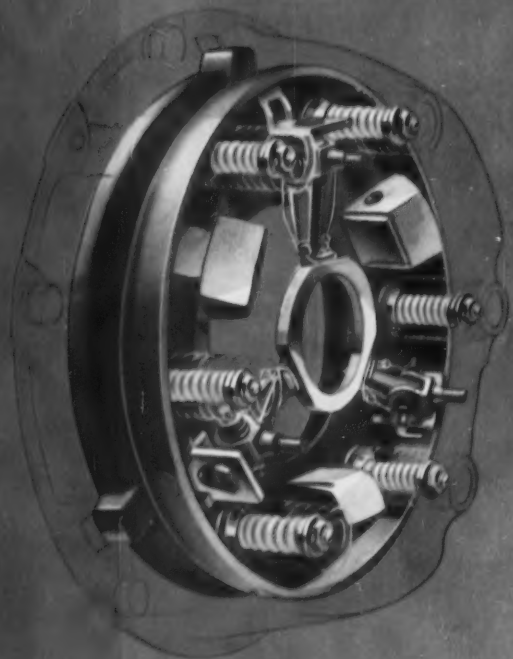
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What causes Steering Heaviness?

To make steering lighter we must know what makes it heavy. It is profitable therefore to start off with a study of the various factors which determine the effort which the driver has to apply to the steering wheel.

We may usefully divide this study into a consideration of parking (or static) and driving (or dynamic) heaviness. Certain factors will be common to both conditions. Let us look at these first. Steering box or gear efficiency is a very obvious one; the road wheel effort plus any further losses are all multiplied up by the inverse of the efficiency to find what the driver must supply.

Friction in all the ball joints is another loss between the driver and the road wheels; so is friction in the idler lever bearings, if any, and more especially in the king-pin bearings.

In parking conditions, however, by far the greatest source of heaviness is the actual torque required to turn the road wheel on its contact patch.

In their paper "The Role of Tyres in Automobile Design and Performance", which was presented at the 1954 F.I.S.I.T.A. Conference in Munich, T. French & V. E. Gough give an expression for this parking torque which is as follows:—

$$T = \frac{\mu W^3}{3 p^{1/2}} \quad \text{where } T \text{ is the torque, } \mu \text{ the coefficient of friction between tyre and road, } W \text{ the load and } p \text{ the tyre inflation pressure.}$$

The derivation of this formula implies centre-point steering; it seems likely that small deviations from this will tend to make the steering heavier, since they will increase the radius of most of the contact patch about its pivoting point. Experimental results which are available show that this parking torque is the highest one which has to be dealt with by the steering.

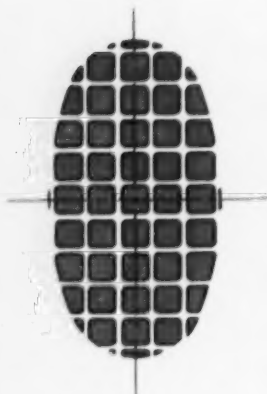
In normal movement on the road, the caster offset of the king-pin axis from the centre of the tyre contact patch implies a torque (positive or negative) which in most conditions is smaller than the self-righting torque of the tyre while it is developing a cornering force. In addition to this self-righting torque of the tyre which has a drift angle, there is a further torque due to the different drags of inner and outer wheels, and their offsets from the king-pin. While a tyre is drifting to develop cornering force, its drag is very considerably increased, partly due to the component of the true sideways force on the tyre acting in the direction of motion of the tyre, and the loads carried by inner and outer wheels are very much different. The drags of inner and outer wheels are also therefore very much different, and the difference in drag multiplied by the king-pin offset provides a torque which has to be overcome by the driver.

These are the major effects concerned in causing steering heaviness; individual consideration of each is necessary to know how they may be reduced.

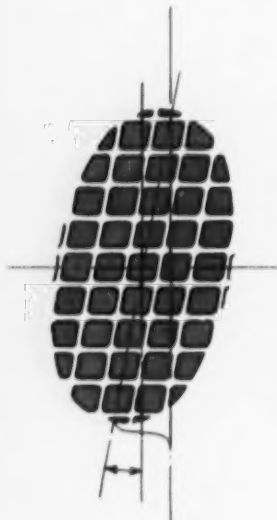
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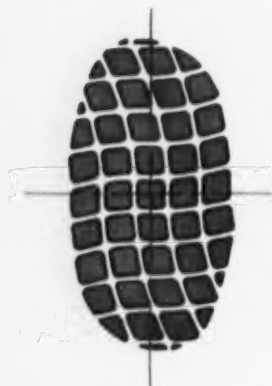
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Undistorted patch



Contact patch with drift angle



Contact patch, parking steering

Taking our own medicine

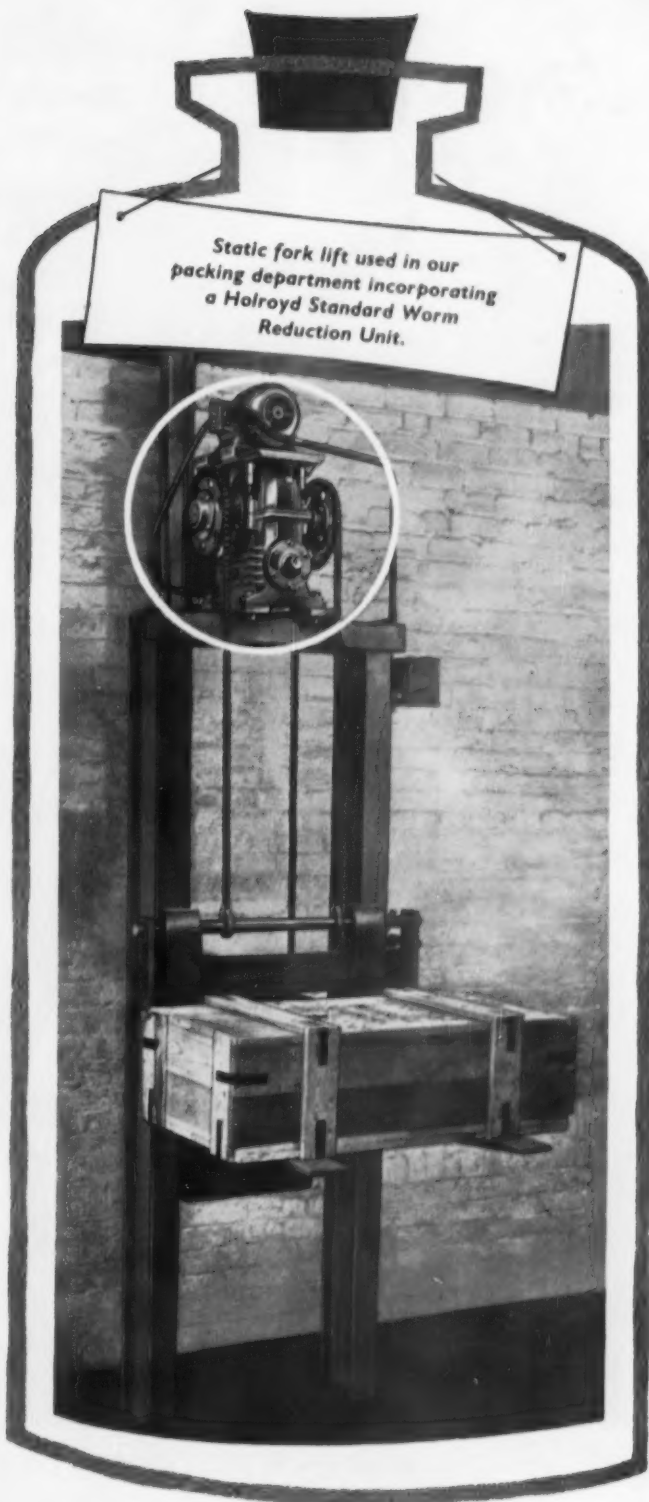
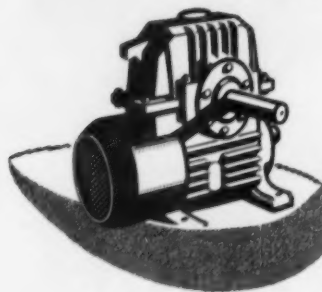
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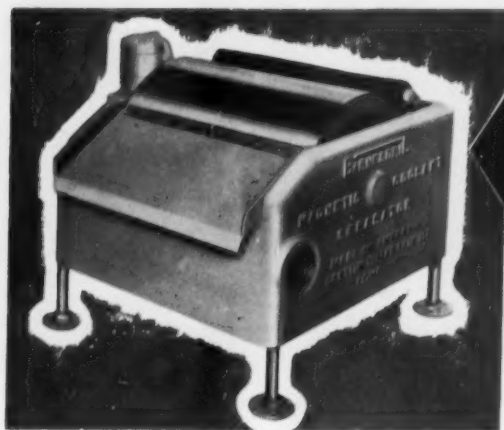
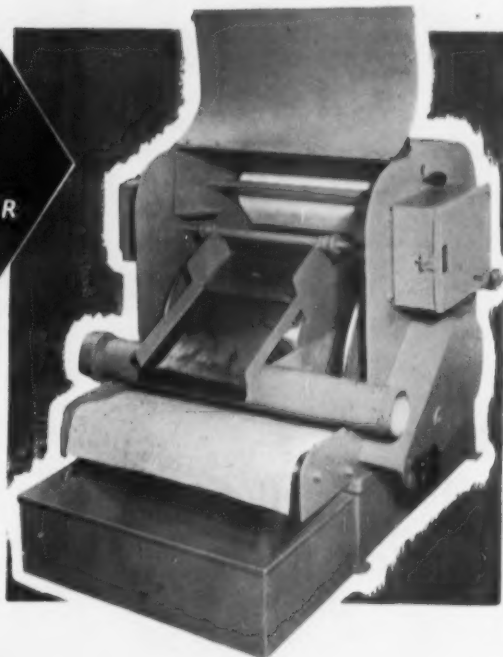
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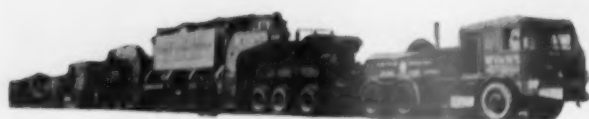
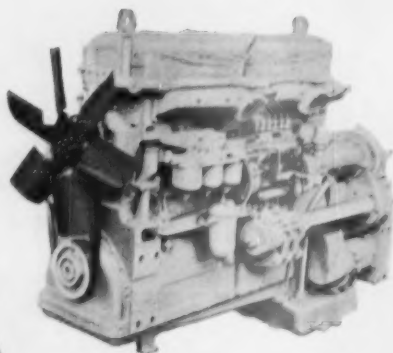
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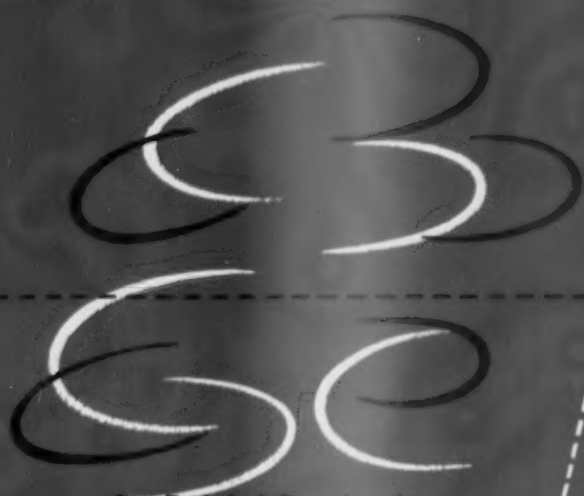
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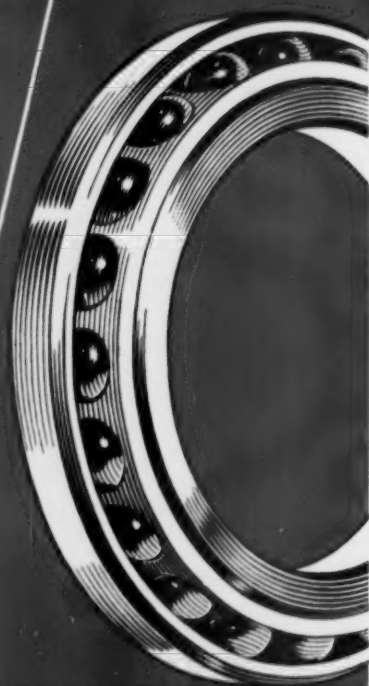
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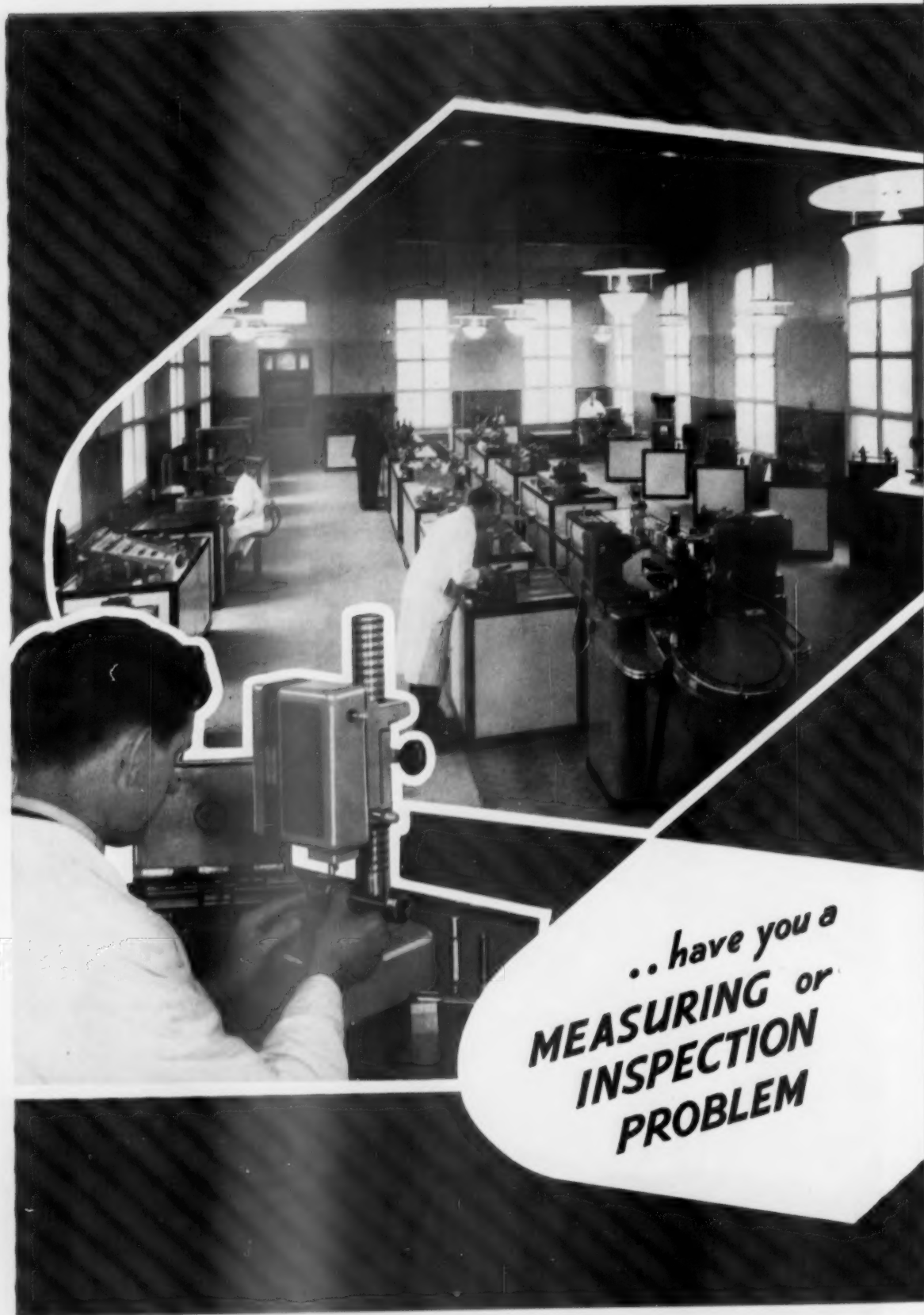
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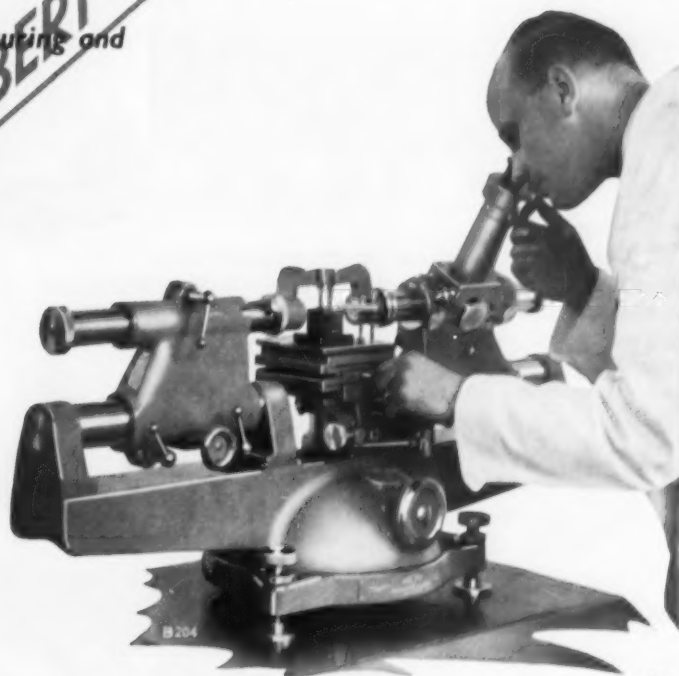
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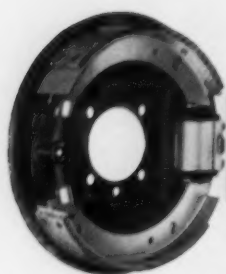
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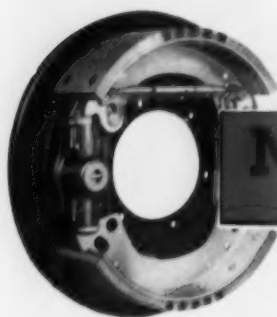


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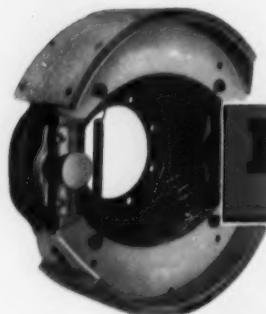
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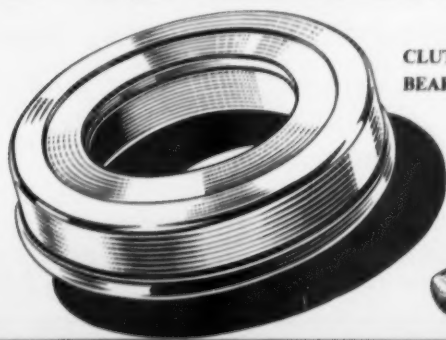


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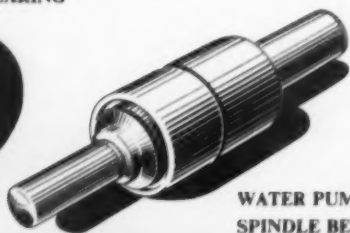


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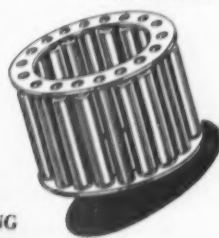
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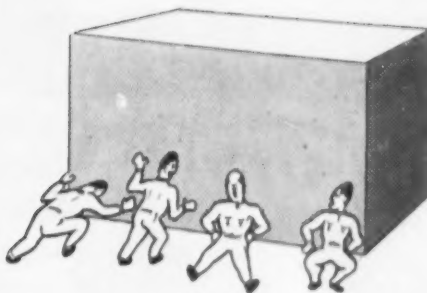
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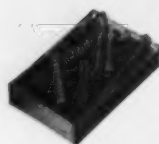


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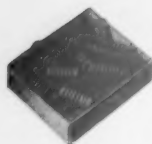
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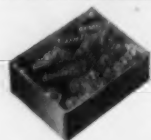
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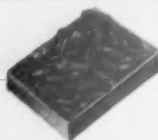
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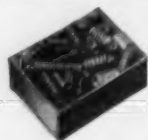
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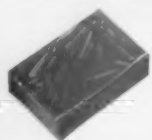
No. 757. Extra Light Compression, 1 gross assorted, $1\frac{1}{2}$ " to $1\frac{3}{4}$ ", $\frac{1}{8}$ " to $2\frac{1}{2}$ " long, 27 to 20 S.W.G. 15/- each.



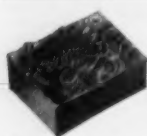
No. 388. $\frac{1}{2}$ gross Assorted Small Expansion Springs. $1\frac{1}{2}$ " to $1\frac{3}{4}$ ", 18G to 21G. 9/6 each.



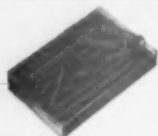
No. 758. Fine Expansion Springs. 1 gross Assorted $1\frac{1}{2}$ " to $1\frac{3}{4}$ ", $\frac{1}{8}$ " to $2\frac{1}{2}$ " long, 27 to 20 S.W.G. 15/- each.



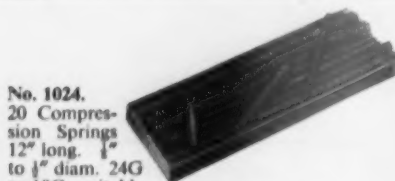
No. 466. $\frac{1}{2}$ gross Assorted Small Expansion Springs $1\frac{1}{2}$ " to $1\frac{3}{4}$ " long, $3/32$ " to $3/16$ " diam., 21G to 24G. 6/6 each.



No. 1013. 1 gross Small Coil Compression Springs, $\frac{1}{8}$ " to $1\frac{1}{4}$ " long, $3/32$ " to $2/16$ " diam., 24G to 19G. 6/- each.



No. 753. 3 doz. Assorted Light Expansion $1\frac{1}{2}$ " to $1\frac{3}{4}$ " diam., $2\frac{1}{2}$ " to 6" long, 22 to 18 S.W.G. 10/6 each.



No. 1024. 20 Compression Springs $12\frac{1}{2}$ " long, $\frac{1}{8}$ " to $\frac{1}{4}$ " diam. 24G to 18G, suitable for cutting into shorter lengths; and 30 Expansions $1\frac{1}{2}$ " to $12\frac{1}{2}$ " long, $5/32$ " to $\frac{1}{4}$ " diam., 22G to 16G. 24/- each.

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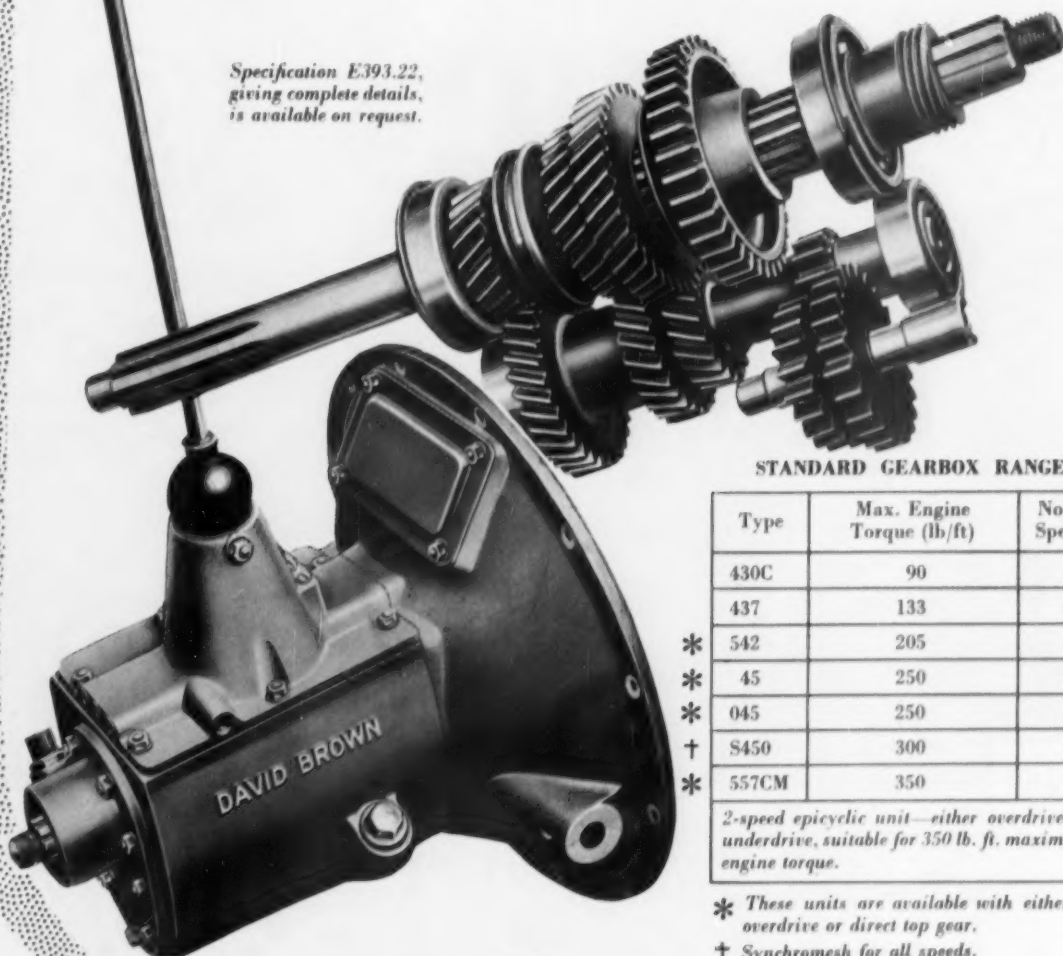
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STANDARD GEARBOX RANGE

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430C	90	4
437	133	4
* 542	205	5
* 45	250	5
* 045	250	5
† S450	300	4
* 557CM	350	5

2-speed epicyclic unit—either overdrive or underdrive, suitable for 350 lb. ft. maximum engine torque.

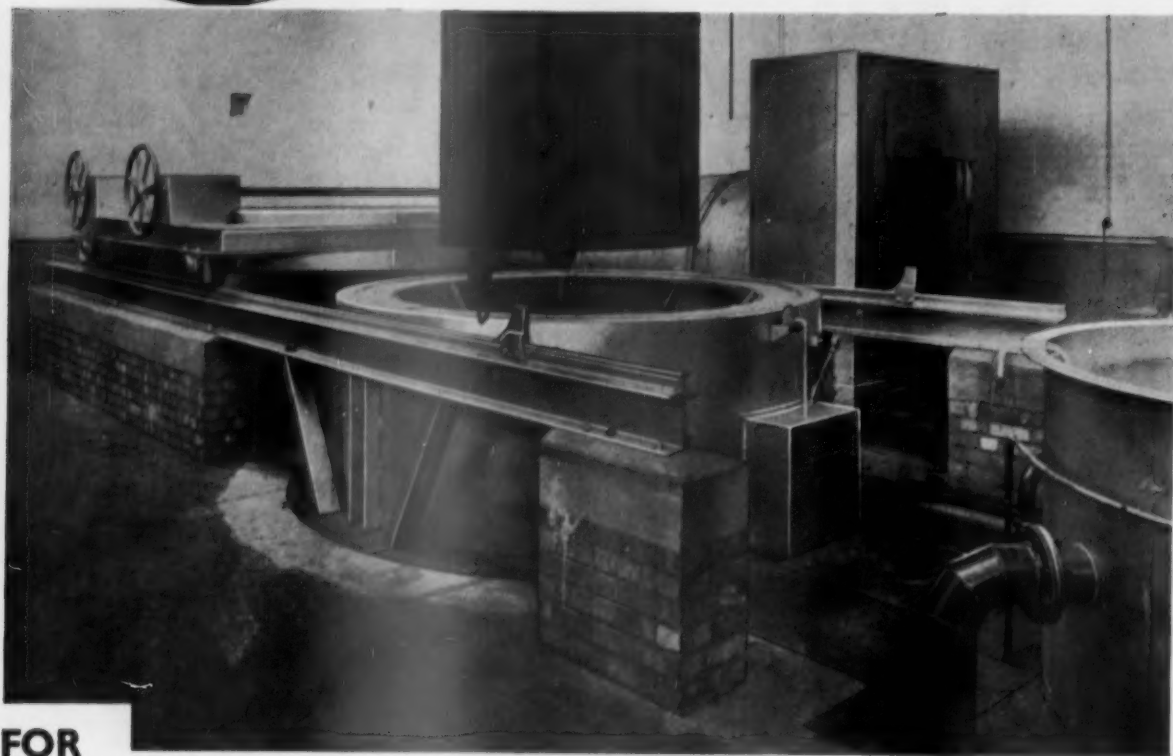
* These units are available with either an overdrive or direct top gear.

† Synchromesh for all speeds.

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DAVID BROWN
 CORPORATION (SALES) LIMITED
 AUTOMOBILE GEARBOX DIVISION
 PARK WORKS HUDDERSFIELD



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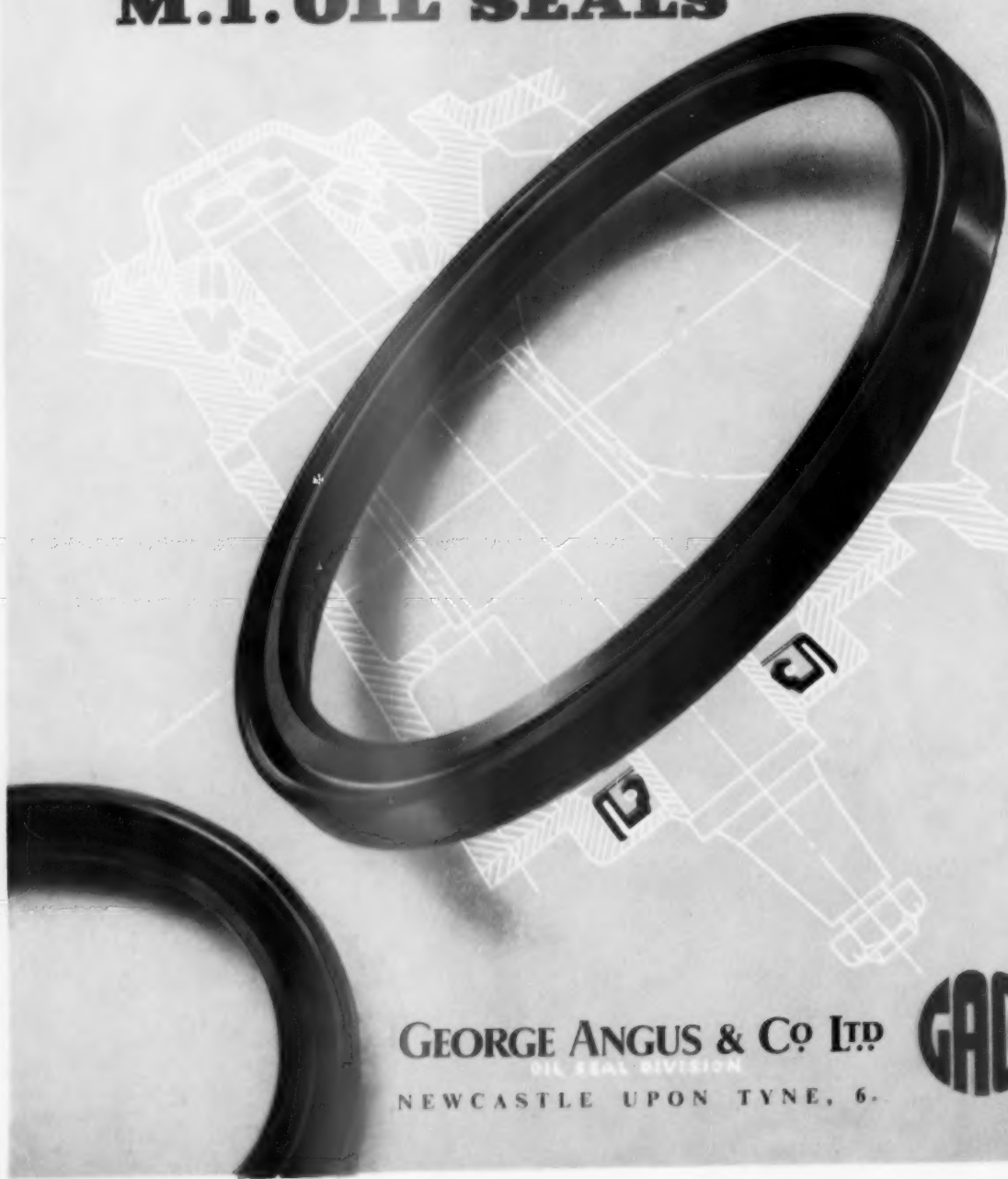


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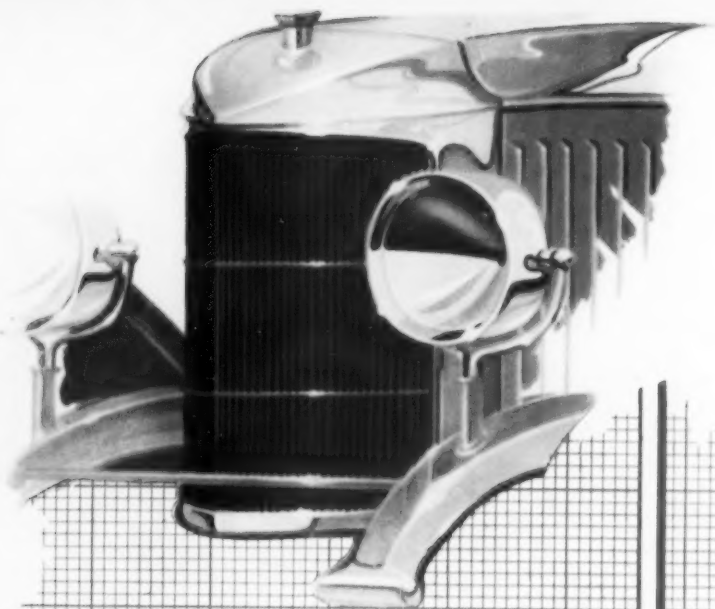


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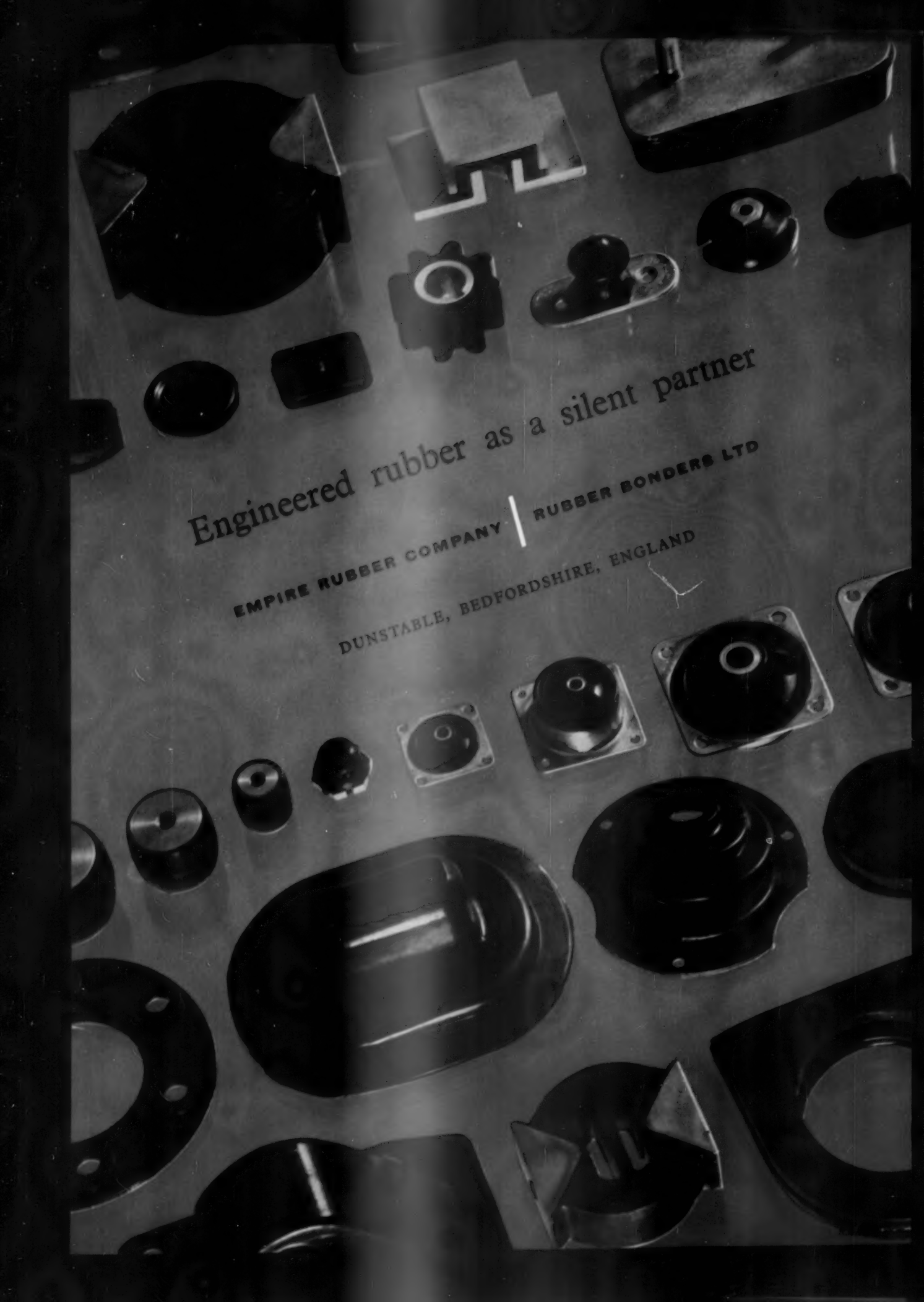


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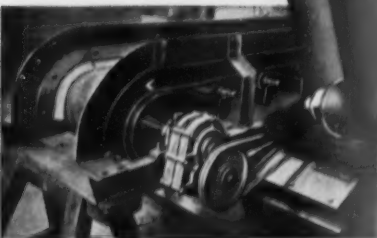
Heavy duty quarry conveyor.



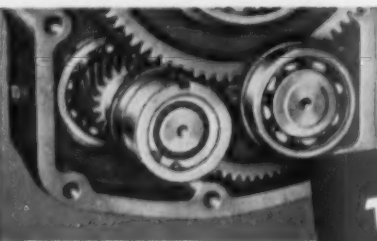
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Back stop to prevent reversal of direction.

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THE LARGEST MANUFACTURERS OF V-BELT DRIVES IN THE COMMONWEALTH

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Soundly constructed, but light in weight, obviating excessive load on the driven shaft and bearings.

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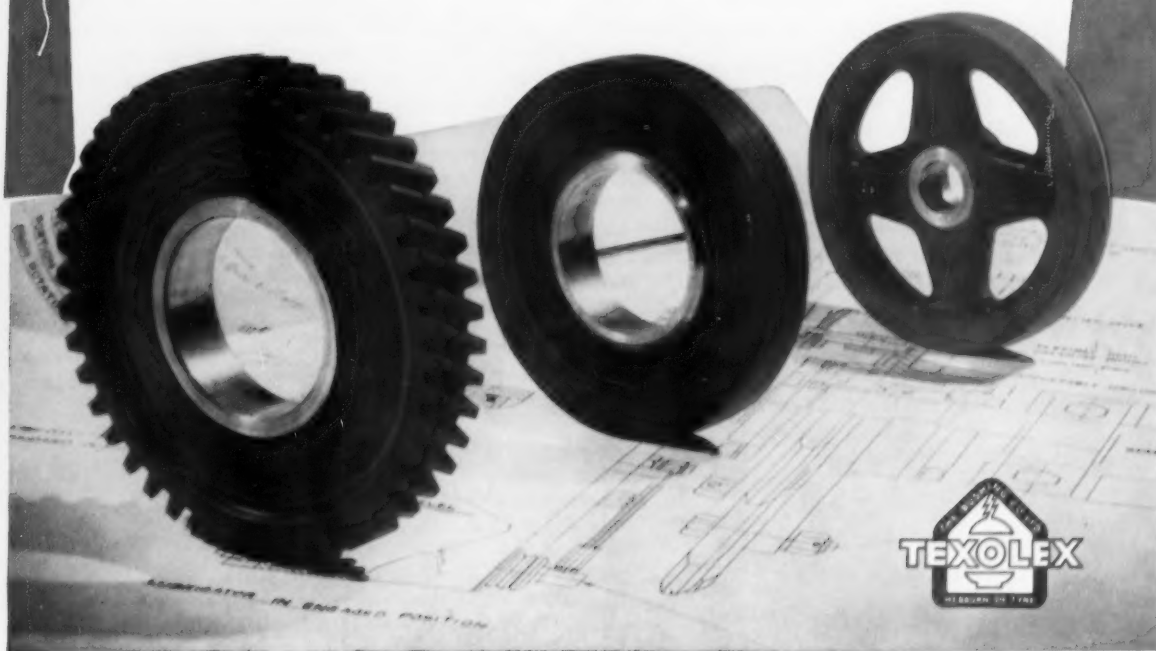


PLASTICS FOR THE ENGINEER

One of the many industrial applications of Texolex laminated mouldings is in the motor industry where extensive use is made of this material for cam gears. Some of the largest motor manufacturing organisations have found in Texolex many of the qualities of the ideal material, offering silence in operation, lightness in weight, high mechanical strength coupled with resistance to wear and corrosion.

TEXOLEX IN THE AUTOMOBILE INDUSTRY

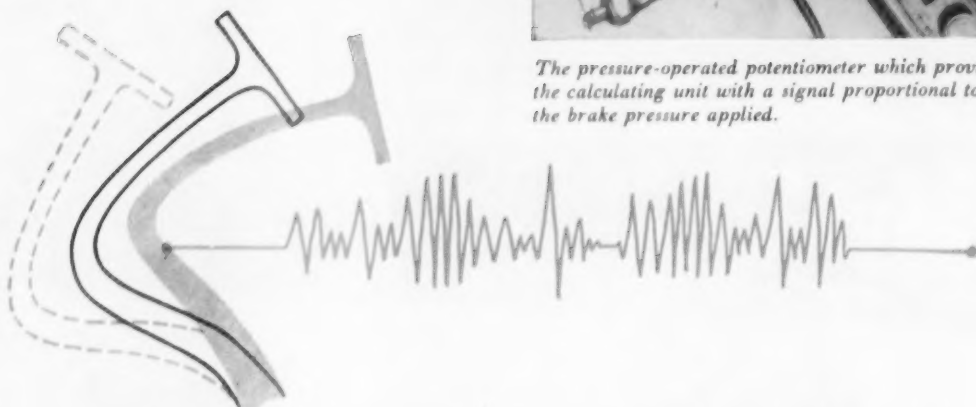
The Bushing Company Limited, would be pleased to give further technical information about Texolex laminated fabric gears and their suitability and operation in camshaft, jackshaft and oil pump drives.



THE BUSHING COMPANY LIMITED • HEBBURN ON TYNE • ENGLAND
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The pressure-operated potentiometer which provides the calculating unit with a signal proportional to the brake pressure applied.



Electronics aid the search for Safety

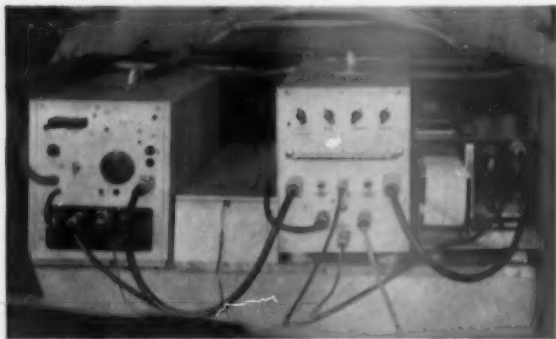
In devising schedules for tests, Ferodo research workers must know what is expected of a brake lining under practical conditions. Without this information it is possible to overrate the lining—make it break down in a manner which would not occur in service.

The performance of a brake lining depends mainly upon the rate at which work is done at the brake and upon the temperature of the drum surface. To record this information Ferodo technicians have installed ingenious apparatus on a vehicle of the Ferodo Test Fleet.

A tachometer generator, driven by the vehicle's propeller shaft provides a signal which is used to record the speed and the deceleration of the

vehicle on a high-speed multi-pen recorder. A voltage proportional to the product of the hydraulic pressure in the brake system and the speed signal is applied to a third pen. Subject to certain necessary precautions, in particular that the brake factor should be frequently checked experimentally, this product is a measurement of the rate of working at the brakes. A thermister is soldered to the surface of the drum to record temperature on a fourth pen.

The results enable Ferodo to devise testing schedules that are accurate and reliable and so to produce brake linings with a high resistance to fade and wear.



FERODO

ANTI-FADE Brake Linings

Some of the apparatus in the back of a test car. The power pack is on the left and the chart can be seen emerging from the four pen recorder.

FERODO LIMITED · CHAPEL-EN-LE-FRITH

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There are certain physical and mechanical requirements expected of every bushing but, provided these can be met there is no need for elaboration. We are bold enough to think (and we have no mean experience) that subject to a small adjustment in the mechanical design, one of our stock cylindrical bronze bearings could often be used where a special type is asked for. As we make both we don't really mind which is going through our machines, but we do know which can sometimes be to the buyer's advantage. Stock sizes really are *in stock*, therefore there is no waiting time for manufacture. Because they are made in far larger quantities the prices are lower. Because they are supplied off-the-shelf they can be bought economically in quantities as low as a hundred or two.

If these factors mean something to you then let us send you details of our standard range. Better still, let us have details of your requirements and our Technical Department will gladly advise.

COMPO

Standard Sizes

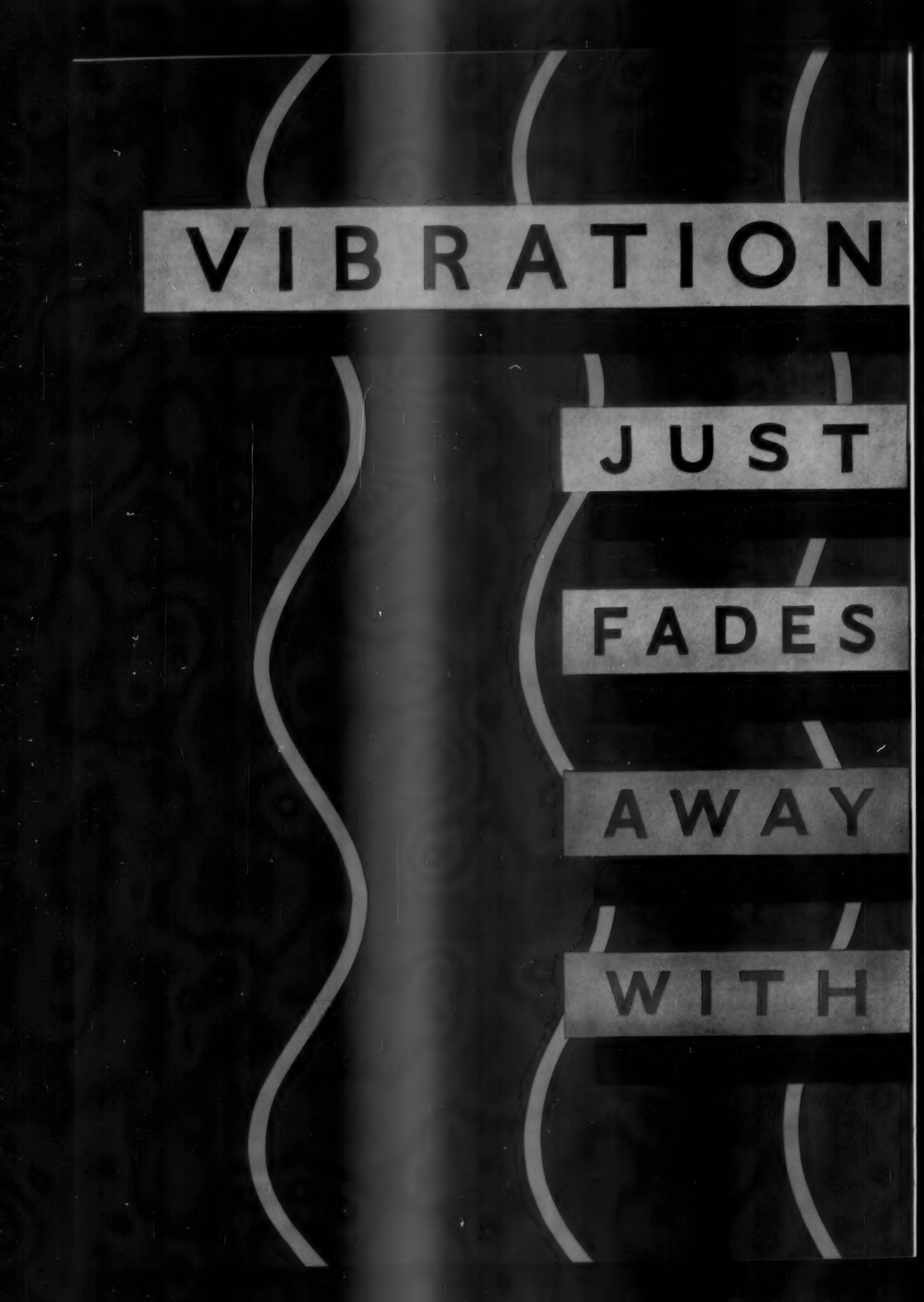
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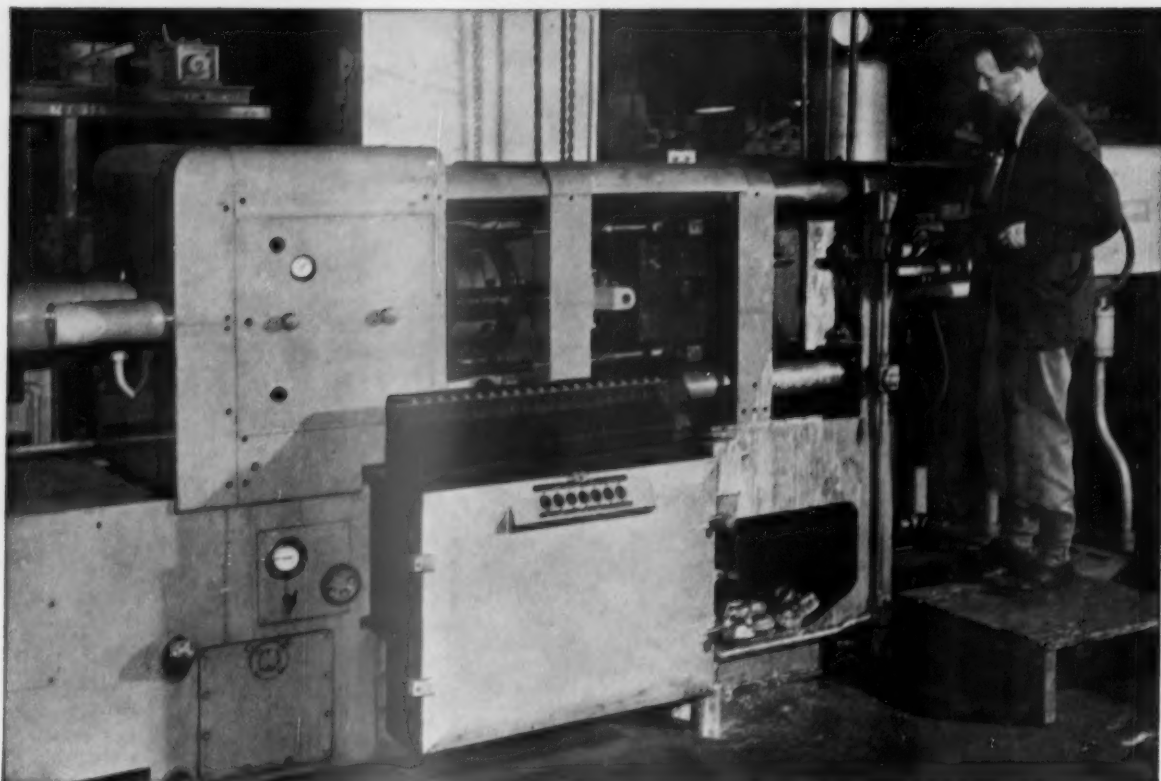
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Reservoil
SINTERED METAL
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Bulldozers are an instance where "Reservoil" bearings are used to advantage. These bearings are supplied in shapes and sizes for every normal medium duty, but attention is drawn to the standard range of plain cylindrical bearings. Write for Catalogue SD 41.

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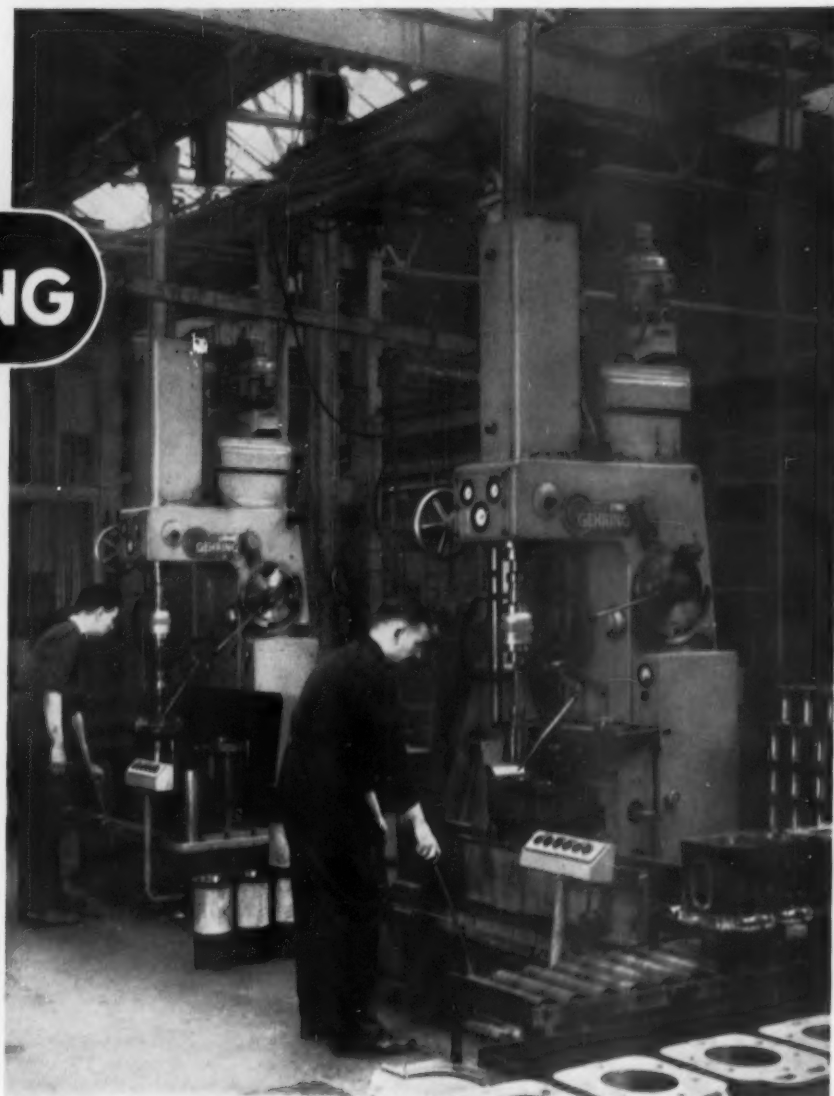
SM 120



HONING MACHINES

For internal
and external
superfinishing

Photograph by courtesy of L. Gardner & Sons Ltd., Patricroft Works, Manchester



Three Gehring machines are installed in the Patricroft Works of L. Gardner & Sons Ltd., the well-known oil engine builders. The right-hand machine in the photograph is used for high precision bore finishing on cylinder blocks, and the left-hand machine is honing cylinder liners.

A further machine is employed for external work, Gudgeon Pins being honed on the outside diameter. A wide range of Gehring machines is available in single and multi-spindle types, and, if required, can be supplied with automatic sizing, whereby the hone retracts when size is reached, before withdrawal from the bore.



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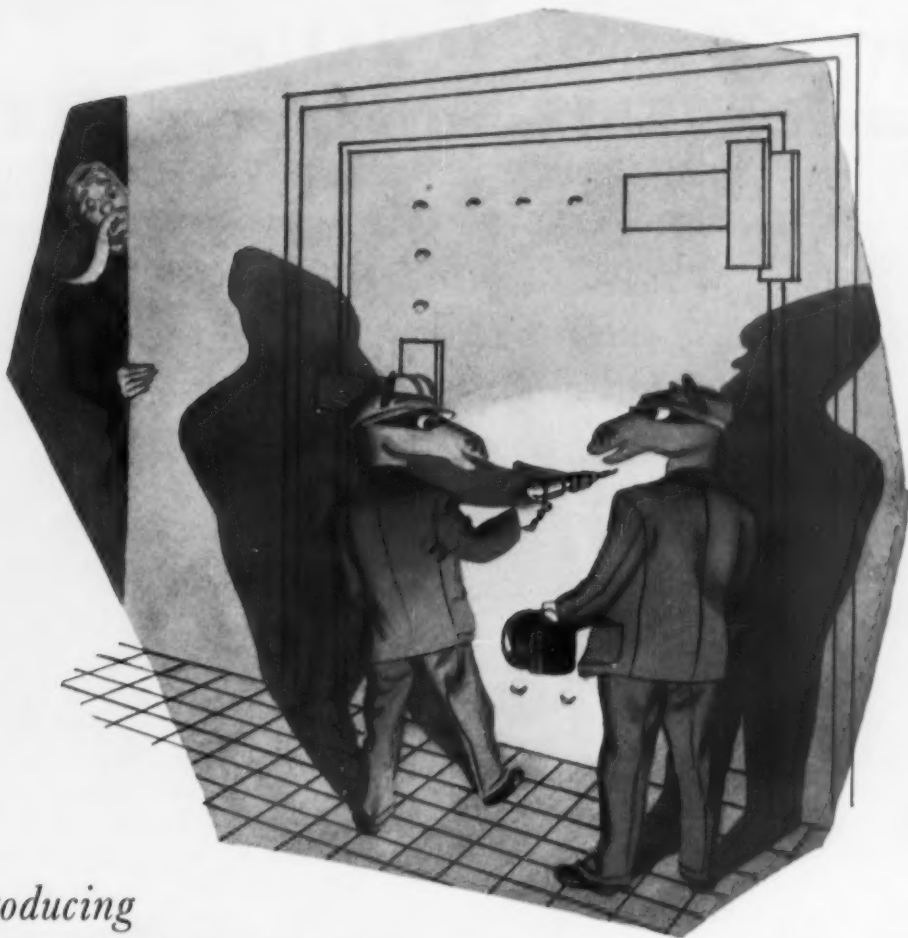
*for drilling up to
4" in steel!*

THIS photograph, taken at the Lincoln Works of Ruston Bucyrus Ltd., shows their wise choice in selecting ARCHDALE for heavy duty vertical drilling machines. Built with a strength and rigidity to match the power required for continuous heavy work, these machines are available in 3 sizes. The 32" machine will drill up to 4" dia. in M.S., the 30" machine 3" dia., and the 28" machine 1½" dia. All three can be supplied with plain, elevating or compound tables.



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People who make safes do so with the idea of frustrating burglars who want to bore their way in. Aircraft Designers appear to have much the same intention towards the long-suffering engineers who have to make and assemble their designs. Take Titanium for instance. (All right, *don't* take it! But if you have to drill holes in an Aircraft Factory you have no choice). Normal high speed power tools work themselves and their operators into a state of great heat, blunt edges and even blunter words. Enter 'Raffles' Desoutter, the Little Horse who works at speeds from 160 to 300 r.p.m. and will bore his way into the Bank of England. He is available in an increasing range of Desoutter Tools.

slow-revving
Desoutter tools

put power into your hands

Here are some of the
DESOUTTER
SLOW-REVVING TOOLS

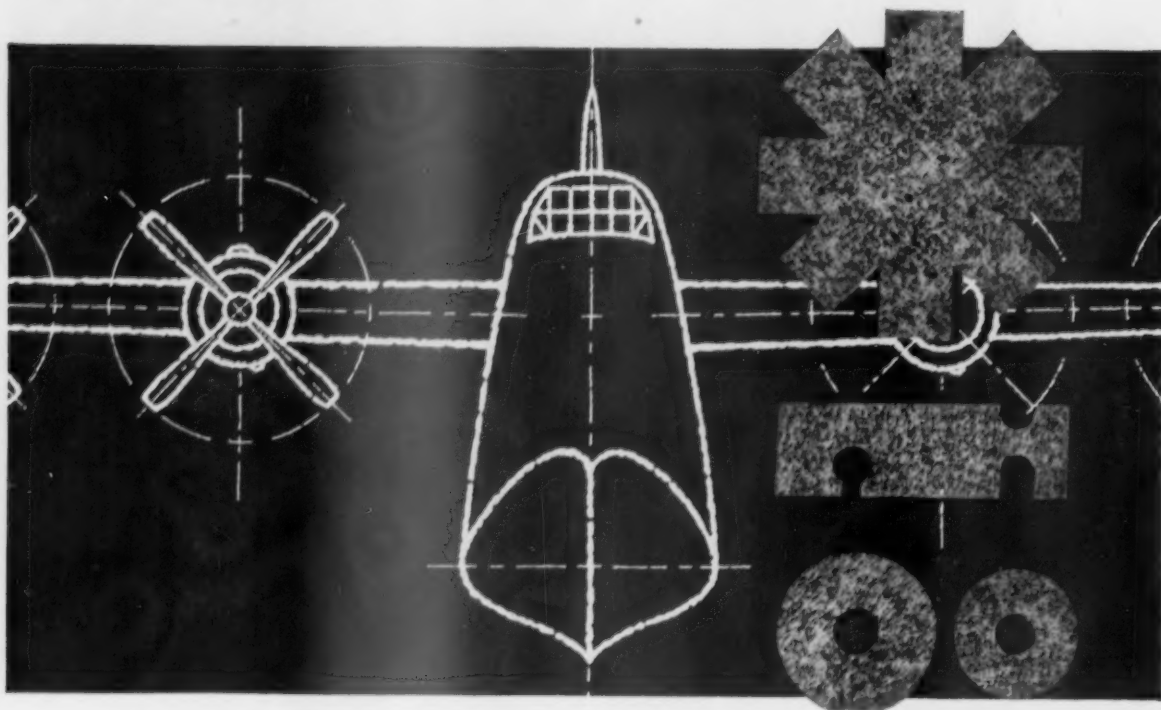
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R39 $\frac{3}{8}$ " capacity drill . . . 165 r.p.m.
SR6/35 $\frac{1}{2}$ " capacity drill . 350 r.p.m.
R141/3 Corner Drill for
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SR10/9 Reamer 90 r.p.m.
Rackfeed Drill (specially
adapted for reaming) . 200 r.p.m.

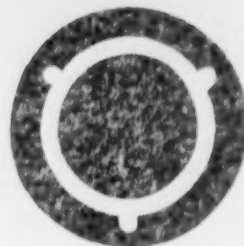
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DG/45 $\frac{5}{16}$ " capacity
Electric Drill Gun . . . 450 r.p.m.

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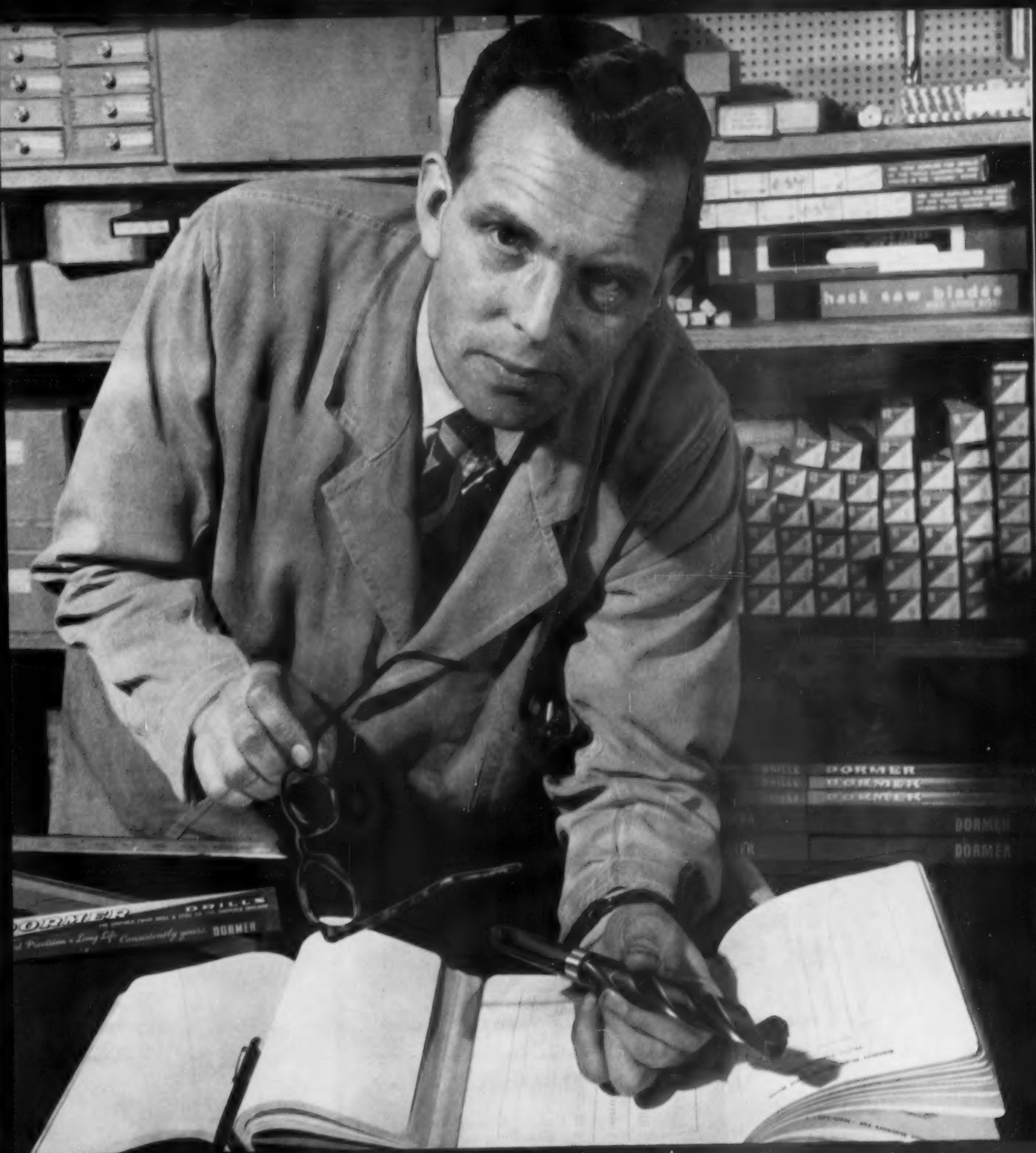


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*it's a good plan
to include*

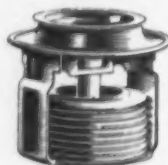


AC OIL FILTERS

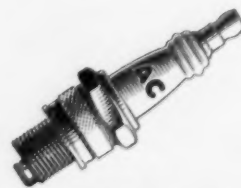
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QUALITY PRODUCTS



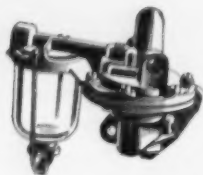
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SILENCERS**



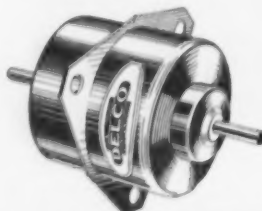
AC THERMOSTATS



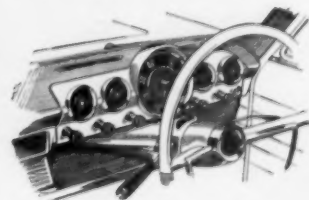
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Pre-lubricated for assembly and running-in

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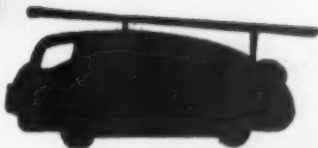
TRUCKS



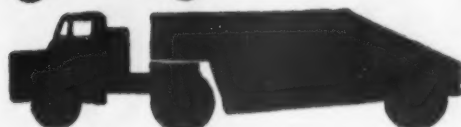
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THE LOAD

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I.C.I. co-operated with the makers in designing the body, and 'Kynal' extrusions and sheets were used throughout.



The spring-lifted composite rear door/ramp is also made from 'Kynal' extrusions.



'KYNAL' ALUMINIUM ALLOYS *Light - Strong - Durable*

IMPERIAL CHEMICAL INDUSTRIES LIMITED, LONDON, S.W.1

M405.A



— OBSERVED FUEL CONSUMPTION TESTS AT SILVERSTONE

16% AVERAGE FUEL SAVING

with 5 different cars fitted with

Laycock Overdrive

COMPARISONS IN M.P.G.

	Without Overdrive	With Overdrive
TRIUMPH TR3 (1991 c.c.)	32.86 <i>m.p.g.</i>	37.46 <i>m.p.g.</i>
VANGUARD III (2088 c.c.)	30.21 <i>m.p.g.</i>	34.69 <i>m.p.g.</i>
HUMBER HAWK (2267 c.c.)	22.84 <i>m.p.g.</i>	27.54 <i>m.p.g.</i>
ROVER "90" (2638 c.c.)	26.01 <i>m.p.g.</i>	31.22 <i>m.p.g.</i>
JENSEN "541" (3993 c.c.)	22.43 <i>m.p.g.</i>	29.04 <i>m.p.g.</i>

Taking an average for the five cars tested with the overdrive unit in operation the reduction in fuel consumption was approximately

16%

16 per cent petrol saving—with the All-British Laycock de Normanville Overdrive in operation—was the surprisingly high average figure gained from a series of R.A.C. observed tests recently carried out at Silverstone with five widely-differing types of cars. The tests, organised by Laycock Engineering Ltd. with the full co-operation of the car-makers concerned, proved conclusively that petrol-saving is high among the many advantages of Laycock "overdrive" motoring.

A line-up of the cars which took part in the tests.



LAYCOCK ENGINEERING LIMITED — SHEFFIELD

Enquiries to: Laycock Engineering Limited, Overdrive and Transmission Sales Division, Westminster Bank Chambers, 16/17 Hertford Street, Coventry.
Under exclusive licence from Auto Transmissions Ltd., Coventry. Electrical controls by Joseph Lucas Electrical Ltd.

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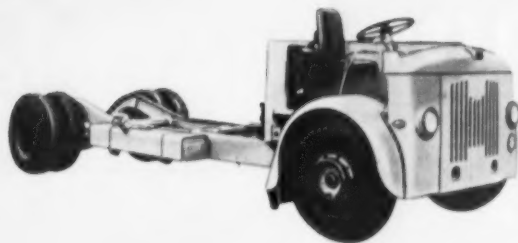
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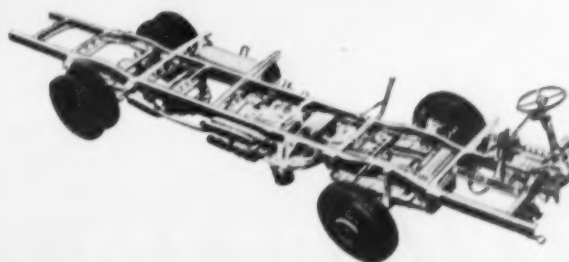
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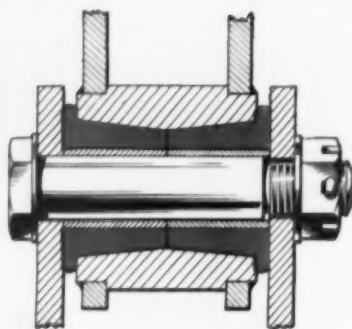
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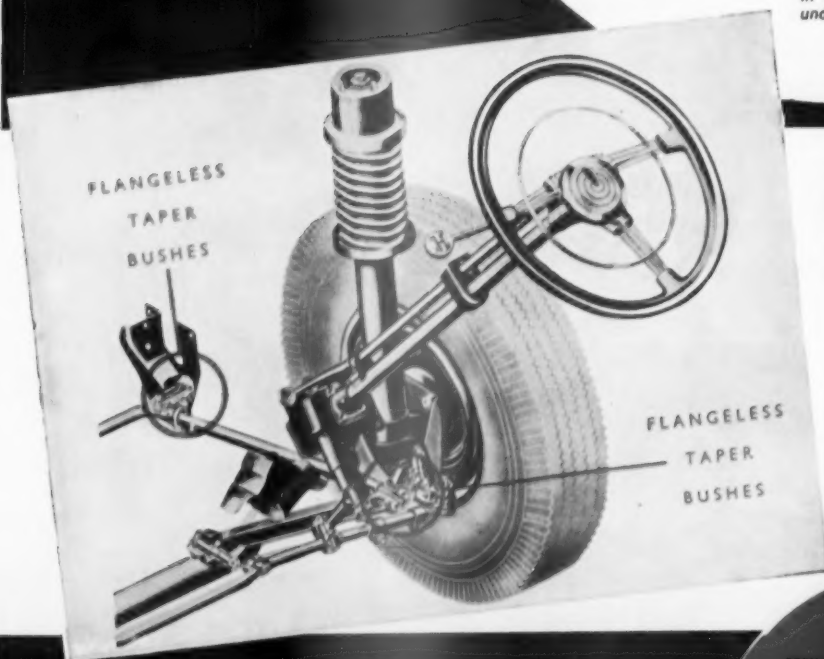


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Right: Flangeless Taper Bush sectioned to show construction. In the housing flanges develop under compression.



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Left: Front Suspension of the "Consul" and "Zephyr Six".

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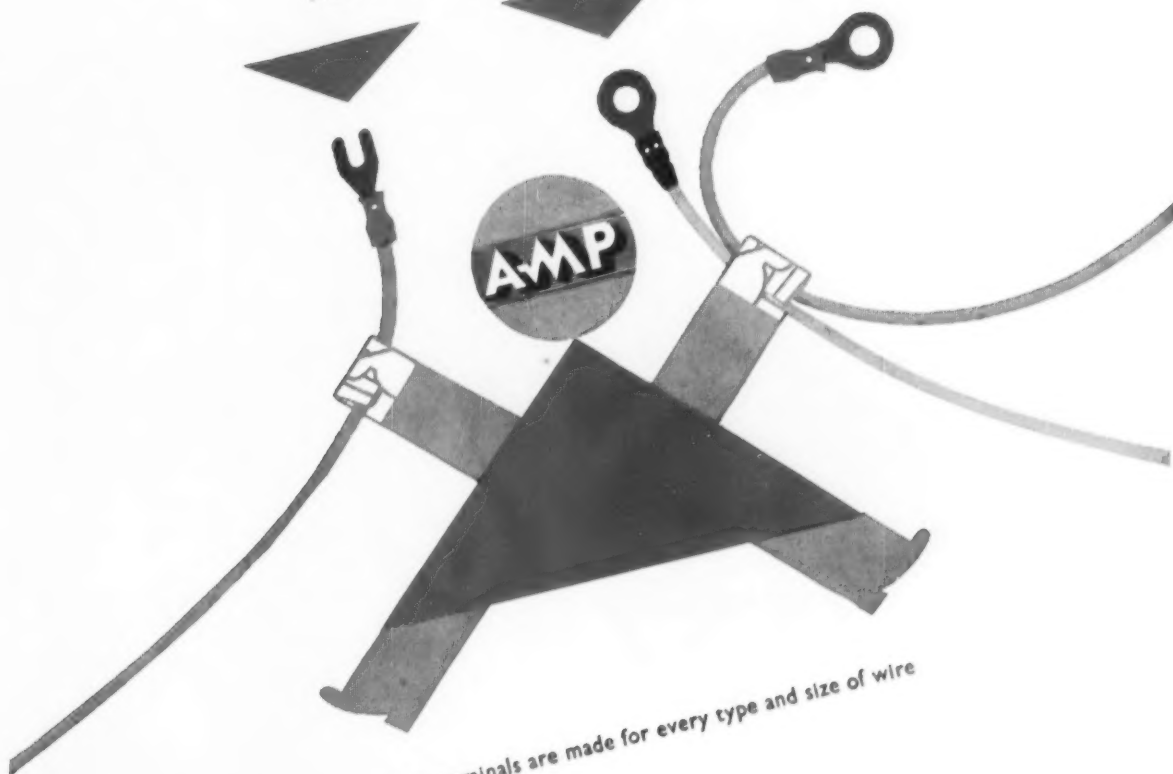
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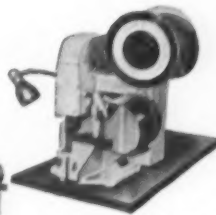
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Illustration, used by kind permission of Messrs. F. W. Perkins Ltd., shows machine consisting of two Delta Milwaukee Hydraulic drill heads mounted on a bed along which the two centring heads operate—used for production location centres at each end of connecting rod. One head carries two centre drills for centring the small ends, the other, 4 drills for centring the large ends of the rods.

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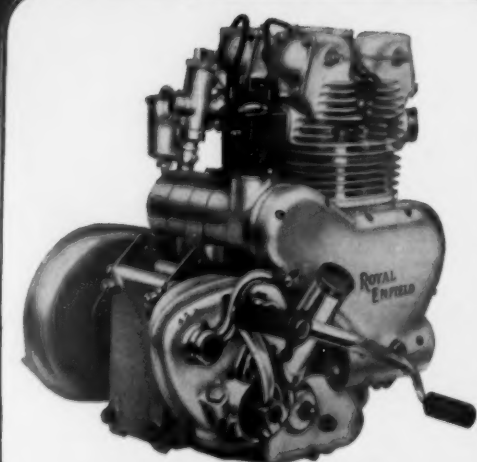
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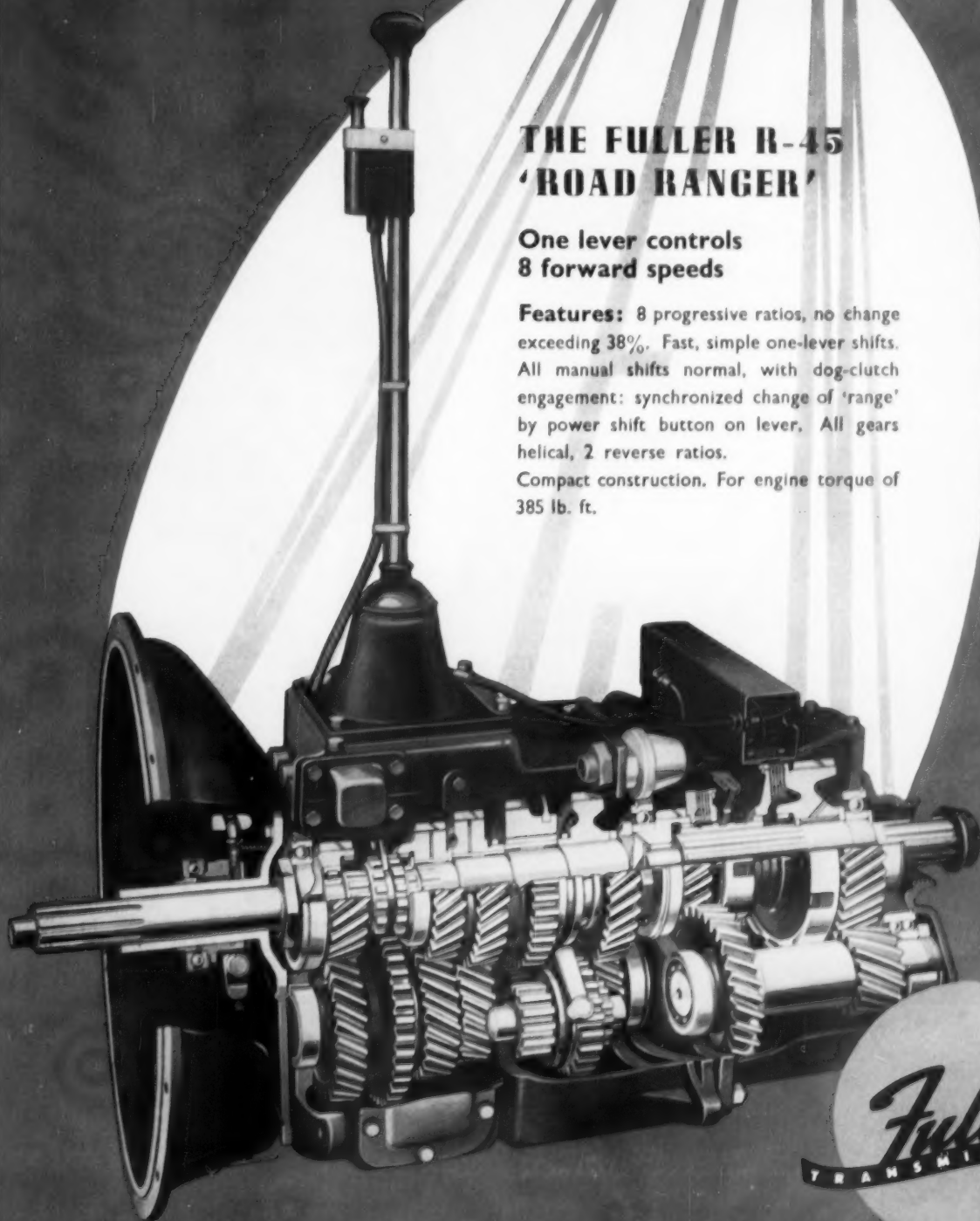
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down

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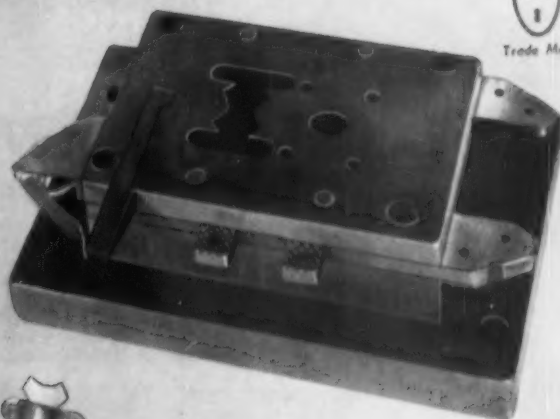
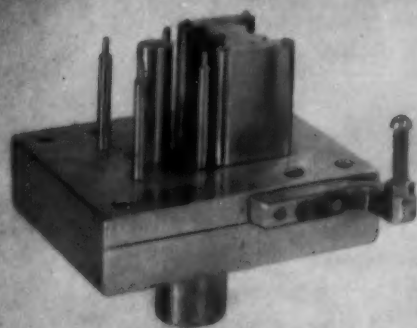
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*The ratio of the motion-resisting force
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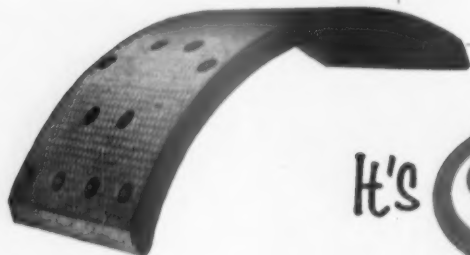
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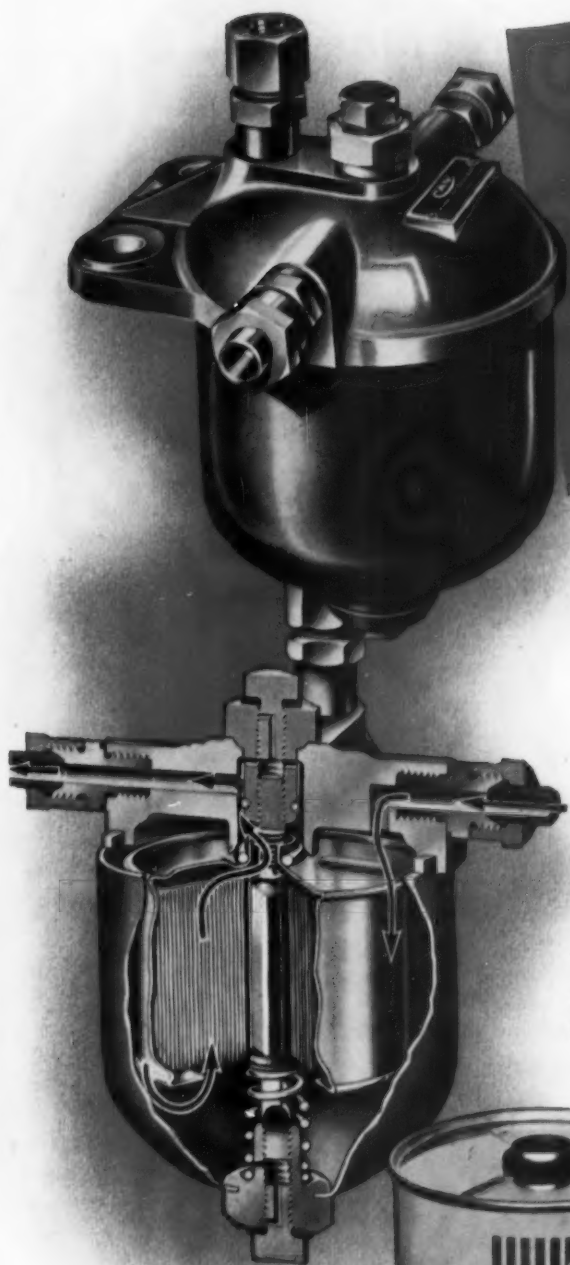
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Filtering surface 8 times area of
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Greatly increases life of
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This new filter has been expressly designed to remove the extremely minute dust particles, carried in suspension in fuel oil, which are responsible for most of the wear in pumps and injectors. Extensive field tests under arduous conditions at home and overseas have proved it possible to obtain eight times the injection equipment life compared with that using older methods of filtration. The overhaul and maintenance costs of fuel injection equipment are thus greatly reduced and engines are kept longer in service.



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FUEL INJECTION EQUIPMENT

C.A.V. LIMITED, ACTON, LONDON, W.3

AP 174/730

**for
casehardening
heat-treatment
brazing . . .**

The 'Cassel' Heat-Treatment Service offers a complete range of salts, furnaces and technical advisory facilities to meet all problems. Surface hardening, for example, is carried out in batch-type or automatic furnaces, using 'Cassel' C.S. 700 or low-strength cyanide-carbonate baths.



A selection of typical surface-hardened components

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- ★ Accurate control of temperature
- ★ Reduced risk of distortion
- ★ Clean, scale-free finish
- ★ Rapid treatment
- ★ Uniform results
- ★ High output for a small floor space

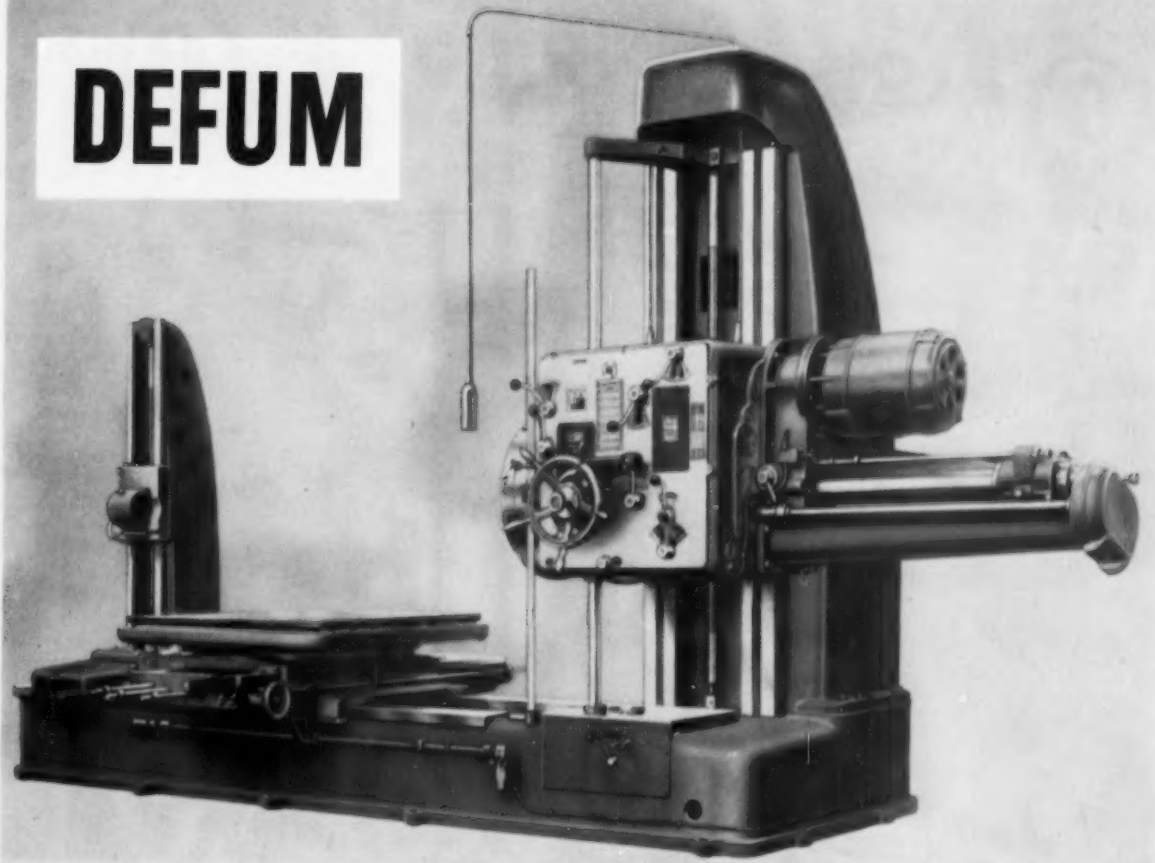
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SPECIFICATION	CWC	HWC
Spindle diameter	80 mm	110 mm
Spindle taper	5 morse	6 morse
Maximum Facing diameter	36"	48"
Maximum boring depth	28"	33"
Maximum distance spindle to table	31"	47"
Maximum distance column to faceplate	7' 3"	9' 10"
Work table	35" x 44"	47" x 55"
Table Feeds:		
Long	39"	61"
Cross	32"	47"
Spindle speeds	(15) 15-515 r.p.m.	(18) 8½-562 r.p.m.
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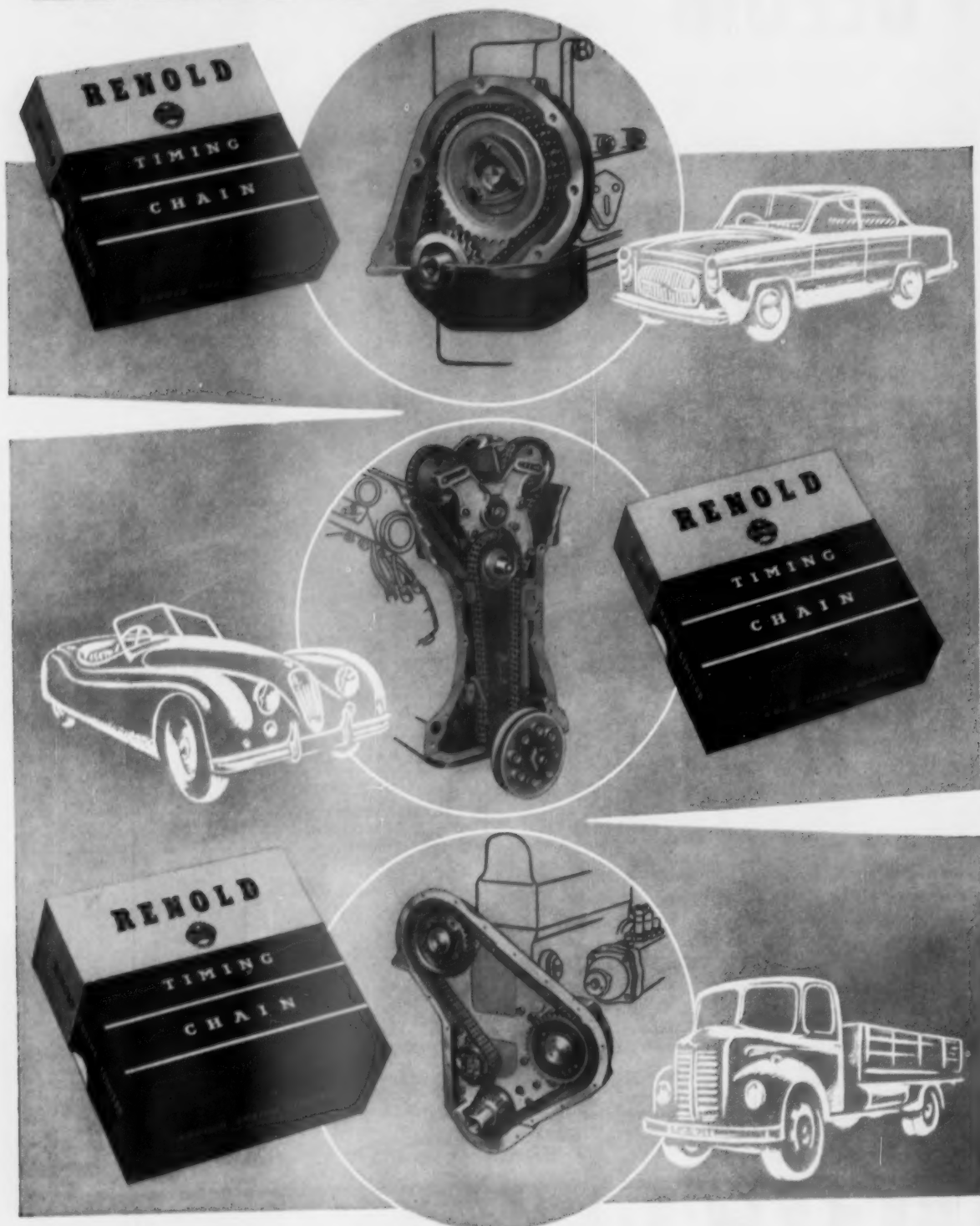
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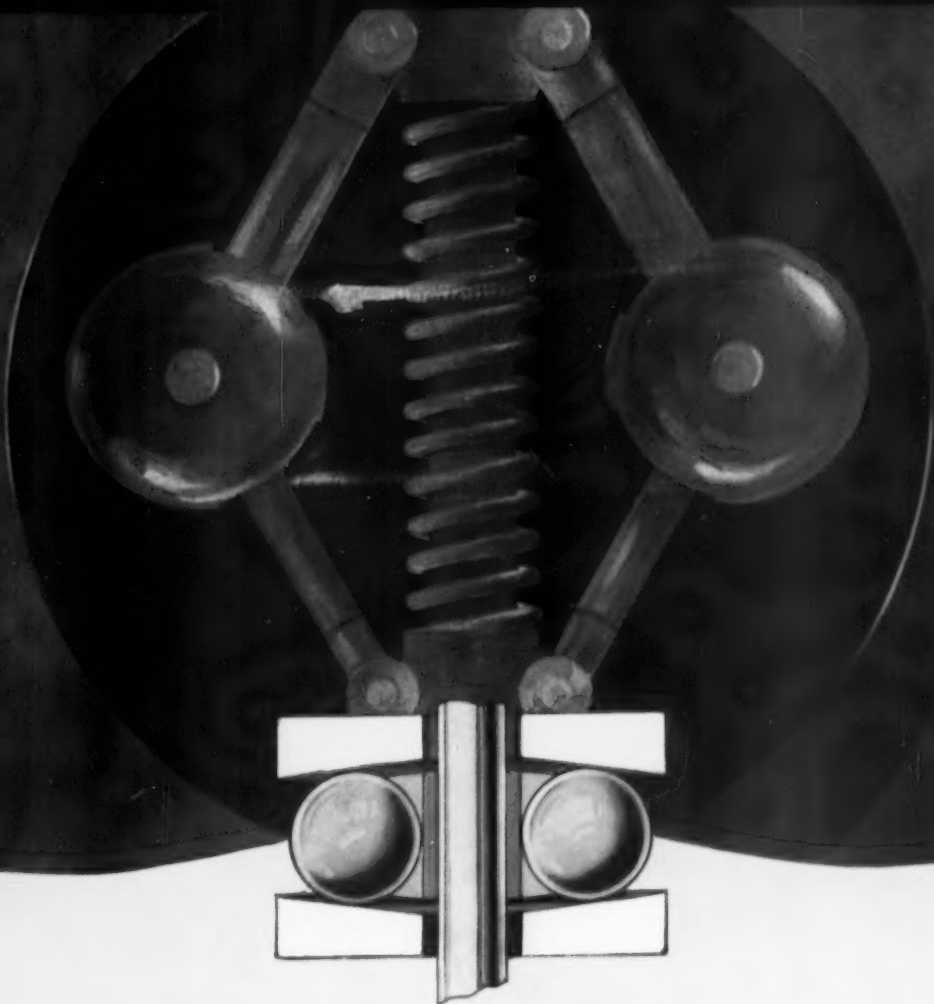
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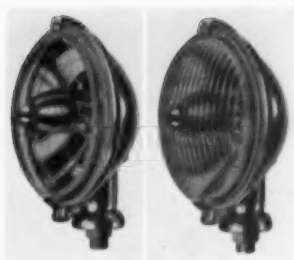
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AUTOMOBILE ENGINEER

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DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

Fiscal Afflictions

ALTHOUGH this journal is normally concerned only with the technicalities of automobile engineering, there are occasions when cognizance must be taken of other than technical factors which affect the efficiency and well-being of the British automobile industry. In the past, for example, strong criticism of the Government's taxation policy for automobiles was completely justified, since the R.A.C. rating formula then used seriously circumscribed engine design development. As is well known, the use of this formula inevitably meant that only high-speed engines of long stroke and small bore were suitable for the popular market. For many years, those responsible for fiscal policy argued that the formula did not have an adverse effect on engine development, but since the change in the taxation system, developments have been concentrated almost wholly on square or over-square engines.

Te-day the British automobile industry is once again in danger of being seriously affected by the Government's fiscal policy. Presumably, the intention behind this policy is the expansion of export trade in vehicles. For the past ten years the automobile industry has earned more foreign currency than any other manufacturing industry in this country, and in 1955, despite increased and very keen competition from other countries and import restrictions in some of the best overseas markets, British vehicle exports were still higher than those of any other country. The maintenance, and if possible the increase, of this supremacy is in very great measure dependent upon price levels. But the measures recently enforced must almost certainly lead to increased costs, which will probably entail price increases.

On more than one occasion it has been stressed in these columns that a healthy home market is absolutely essential if export markets are to be retained and expanded for cars in the popular ranges. This is not to suggest that the home market will, or should, compete with export markets. What it does mean, however, is that when for any reasons exports fall off, the home demand must be sufficient to allow full production to be maintained. Unfortunately, this does not seem to be realized in official circles.

It may, of course, be the official view that a restriction on home sales will lead to greater sales efforts in other markets, but this is unrealistic. During the past ten years unsurpassed efforts have been made to expand existing and open new overseas markets for British automobiles. It is

difficult to see how reducing the potential home demand by increasing purchase tax and making hire purchase more difficult will lead to any appreciable increase in the great efforts already being made to sell vehicles overseas. It is, however, all too probable that these measures will lead to cuts in production (several have already occurred) and consequent increases in production costs. Any such increases can, of course, be met in one of two ways; either by a rise in selling prices or by a reduction in profit margins. Neither holds out much hope for the future.

A rise in selling prices will cause a further drop in sales, followed by further cuts in production and further rises in manufacturing costs, and so *ad infinitum*. Nor is there any reason for thinking that a reduction in profit margins would be in the interests of the industry or the country. In fact, such a policy would almost certainly lead to a reduction in the funds available for capital investment to maintain factories at the highest possible level of efficiency. There is also a probability that reduced profit margins would have an adverse effect on the funds available for design development, an item which really needs increased, not diminished, expenditure.

In this connection, some figures quoted by *The Financial Times* regarding the German automobile industry, this country's greatest competitor, are of interest. If the price index for 1950 is taken as 100, the index for German cars is now 99. Yet during the period 1950-1955, German raw material prices have risen sharply, sheet steel for example, has risen by 80 per cent, and wage rates have increased by 40 per cent. That German motor cars are now on the average cheaper than in 1950, though superior in quality, is attributed mainly to the manufacturing economies consequent upon greatly increased production. Although, as we in this country know only too well, the German vehicles have made successful incursions into export markets, the leaders of the German automobile industry still believe that its strength rests on a firm foundation of expanding home demand.

We do not object to these fiscal afflictions because of any wish to have the automobile industry protected against any economic ills that may beset the country. Our objection stems from the fact that we sincerely believe that they will tend to prevent the industry from playing its full and vital part in the development of a balanced economy free from the stresses and strains of recent years.

Girling Power-Assisted Steering

A Unit With an Ingenious Valve Spring Arrangement

THE Girling power-assisted steering arrangement was demonstrated to the public for the first time at the 1954 Earls Court Exhibition. This manufacturer states that interest in the system is so widespread that it is difficult to cope with all the enquiries that are continually being received. Two units are currently being offered: one is the 1½ in bore steering unit for cars and light vans, while the other is a 2½ in bore unit for heavier vehicles. The smaller one gives a thrust of 1,075 lb, while the larger unit gives 2,400 lb thrust. No application has yet been found that is too heavy for the larger unit. However, this might be because the system so far has been applied to vehicles initially designed for operation without power assistance and which therefore are reasonably light to steer.

Both units are served by a Hobourn Eaton, engine-driven pump, shown in Fig. 1. If necessary, the pump and reservoir can be supplied separately, so that the reservoir can be mounted on the vehicle structure instead of on the pump body. One size of pump has been found adequate for both the small and large steering units. Its output is sufficient if it is rotating at as little as 700 r.p.m., and its maximum speed

unit and the steering unit, and one for the return to the reservoir. Reduction of the number of hoses not only lowers the cost, but also decreases the vulnerability of the system to accidental damage. Moreover, the larger the number of flexible hoses, the greater is the lost motion and sponginess due to expansion of the system under hydraulic pressure. When the two-piece system is used, the valve unit is generally incorporated in the drag link which, therefore, must be a special component, whereas in the one-piece arrangement the drag link only has to be modified by shortening it, in some applications by as little as 1½ in. Employment of the two-piece arrangement also involves the fitting of two brackets to the steering unit instead of one. In view of the obvious advantages of the one-piece type of unit, this layout has been adopted by Girling. Nevertheless, should a sufficiently large demand arise for the two-piece type for a special application, no doubt it would be met.

Both the Girling units have been designed so that a minimum amount of modification is needed to existing vehicles to adapt them for power steering. One end of the steering unit is anchored to the vehicle structure, on which a suitable bracket is, of course, required. At the other end are the two connections to the steering drop arm and the drag link. One of these is a special ball pin fitted to the drop arm, and the other is the ball joint on the end of the drag link. As already mentioned, this link has to be shorter than if it were connected directly to the drop arm.

The accompanying illustrations, Figs. 2 and 3, of the 1½ and 2½ in bore units are representative of the type of layout that is offered, but they are experimental arrangements and no doubt will be varied slightly to suit different installations. Two different types of unit are available in each of these two sizes. With one, the valve unit is arranged in line with the drag rod, but with the actuating unit alongside; in the other, the drag rod, valve and actuating units are all in line on a common axis. The essential components of both types are similar, except in that the valve unit of the type that is not in line must have the device to prevent any tendency for it and the drag rod assembly to rotate round the jack, or vice versa, whereas the in-line type might be used without this device. In general, the moving parts of each type of valve unit are common to all applications of that kind. Thus, the prime cost is kept to a minimum and spares will be readily available. The in-line version of the smaller unit and the side-by-side arrangement of the larger one are shown in the accompanying illustrations.

By using one or other of these alternative types, it is possible to install the system, without altering the steering geometry, in any existing design of vehicle which incorporates a beam axle and in most with independent front suspensions. When the in-line type of unit is employed, it either has a rod extension that forms the drag link, or it replaces the drag link and has lugs attached to its body to receive the ends of the steering rods. If the other type of unit is employed, the drag link is attached to an extension of the valve body.

The in-line installation, of course, is the more compact one. Several types of ball end fittings are suitable for the attachment of the piston rod to the frame; only one of these is shown in the illustration. The piston rod is ground to a finish of 10 micro-in and chromium plated to give good wearing qualities. A square section, rubber ring is housed in the outer end of the cover that closes the end of the jack; it acts as a wiper to prevent abrasive matter entering the

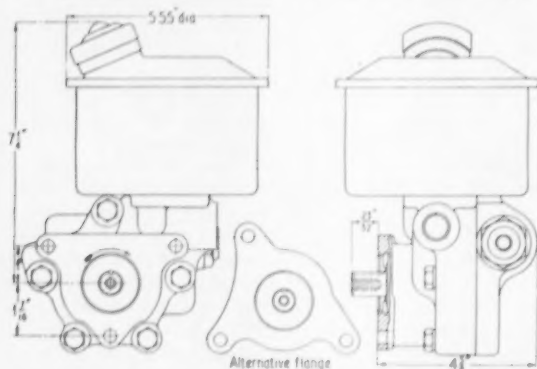


Fig. 1. The Hobourn-Eaton pump employed in the Girling power-assisted steering systems can be installed either as shown in this illustration, or with the reservoir mounted separately

should be limited to 6,000 r.p.m. The Hobourn Eaton pump was chosen because the output of the standard unit currently in production is suitable for this application, the unit is reliable, relatively inexpensive and has a long service life. S.A.E. 10W oil is recommended for the system.

One of the first decisions that Girling had to take was whether to manufacture a two-piece or a one-piece steering assembly. In the two-piece arrangement the valve and the steering units are separate, whereas with the one-piece layout they are not. Incidentally, the Americans include the pump as one of the pieces and call these the three-piece and two-piece arrangements respectively. Although the two-piece arrangement (by the British definition) may in some instances be the less expensive so far as the steering unit is concerned, the fact that additional components are needed to install each piece separately, generally more than offsets this economy.

With the two-piece layout there are inevitably four flexible hoses and their end fittings, whereas the one-piece type of unit requires only two. The four hoses are: one between the pump and the valve unit, two between the valve

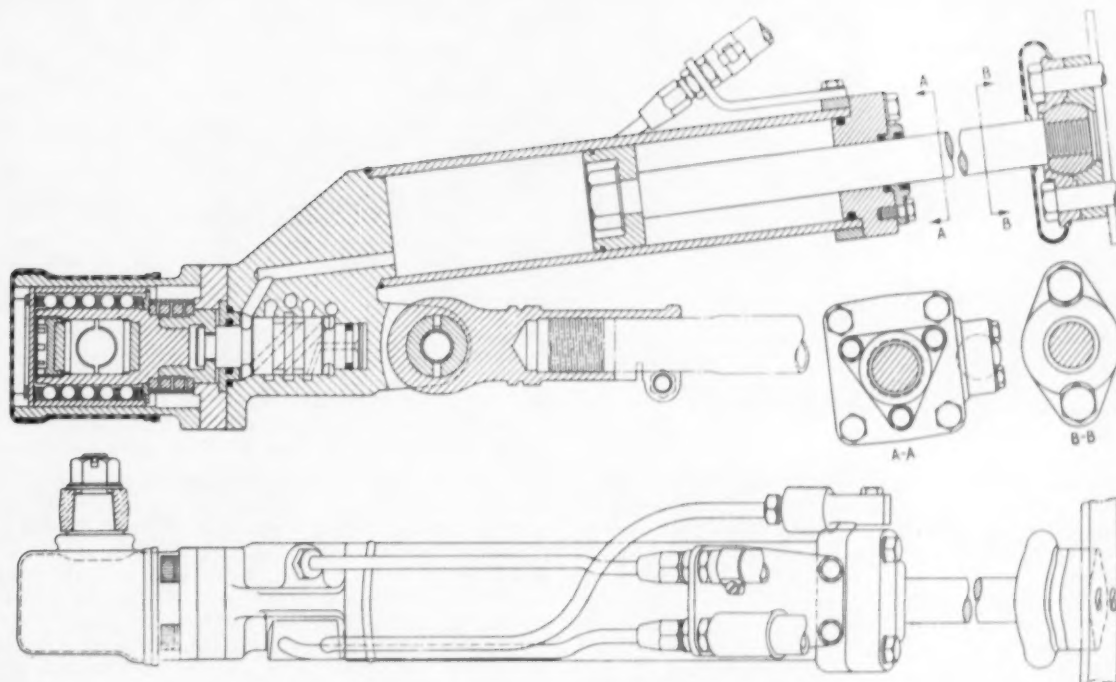


Fig. 3. For some applications, it is more convenient for the valve unit not to be in line with the jack, as in this arrangement. In this $2\frac{1}{4}$ in bore unit, square section wire is used for the centralizing spring

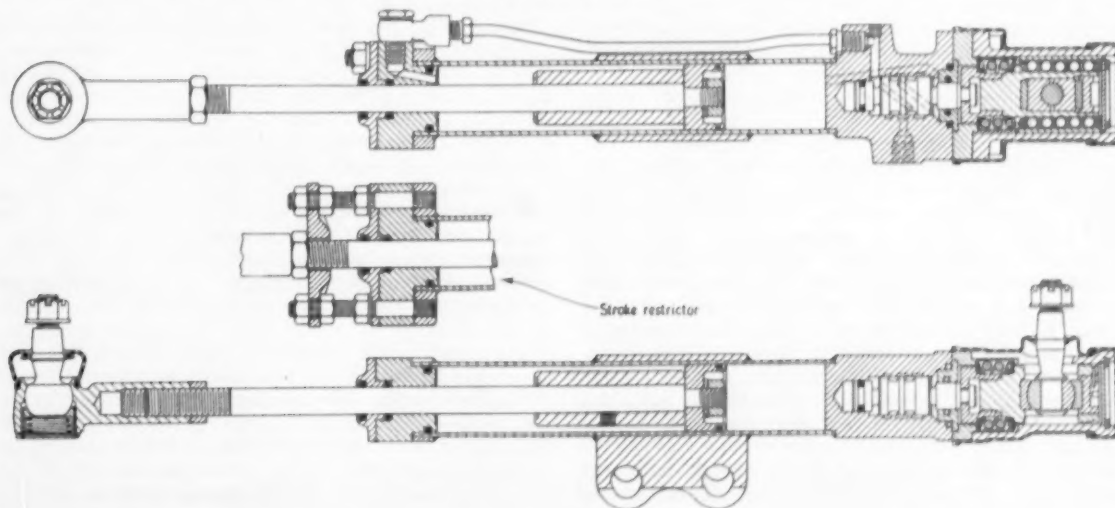
bearing that carries the rod. The diametral clearance between the rod and the bearing is in the order of 0.0005-0.0020 in.

Two high pressure seals, in the form of circular section rubber rings, are employed at this end of the unit. One is housed in a counterbore in the outer end of the bearing, where it is retained by the spigot of the end cover, which projects into the counterbore. The other ring is housed in a groove round the outer periphery of the inner end of the bearing, which is spigoted into the cylinder. A banjo union, for the pipe that carries both the high pressure oil supply to and return from this end of the jack, is screwed radially into the periphery of the bearing. It communicates with an

oilway drilled from the inner end face of the bearing. The other end of the pipe is connected to the valve body. Spigoted on to the end of the rod is the aluminium alloy piston. It carries a conventional, cast iron, piston ring type seal in a groove round its outer periphery.

In the experimental unit illustrated, the valve body is spigoted into the other end of the cylinder, but the possibility of making these two components integral is being investigated. The sliding component of the valve is made in three pieces. One is an Alford and Alder ball joint, which is self-adjusting for wear; the ball pin of this joint is carried in the lower end of the drop arm of the steering box. The centre component is the valve spring retainer, which is screwed on to the ball

Fig. 2. In the Girling power-assisted steering unit with the valve in line with the jack, a single spring made of circular section wire is employed to centralize the valve



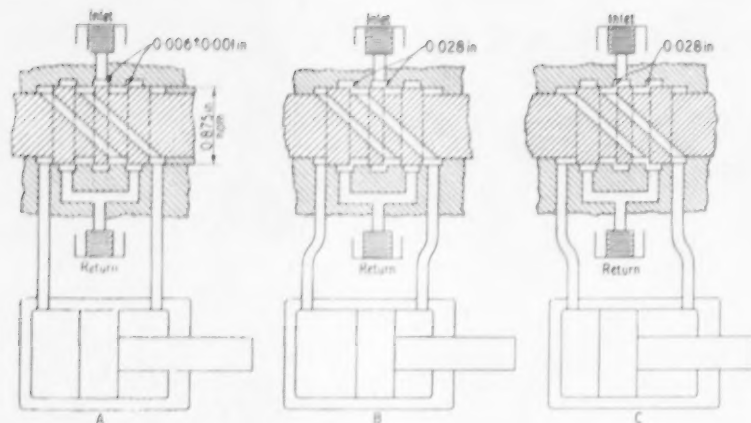


Fig. 4. Diagrammatic illustration of the operation of the slide valve

- A Valve in neutral position, power cylinder not operating
- B Valve slide moved to left, power cylinder extending
- C Valve slide moved to right, power cylinder retracting

joint cup, and the other is the valve bobbin. Assembly of the valve bobbin to the spring retainer is effected simply by passing the end of the bobbin, round which is a peripheral groove, into the circular portion of a key-hole slot machined radially in the end of the retainer, and then sliding the bobbin into the parallel-sided portion of the slot until both components are coaxial. In this position, the sides of the slot register in the groove round the end of the bobbin; close tolerances are maintained in the manufacture of these components, to provide positive axial location.

The valve body also is in three pieces. One is spigoted into the end of the cylinder of the steering unit and forms the housing for the valve. In the intermediate component, the valve spring is housed. The third component carries the ball joint that connects the unit to the drop arm; its outer end is cupped to receive the ball joint on the end of the drag rod.

Both ends of the bobbin form reaction pistons, whose function will be explained later. The end nearest the cylinder is grooved peripherally to receive a circular section rubber sealing ring and is carried in a hole in the valve housing. A separate bearing, flanged and clamped between the spring and valve housings, carries the other end. The seal round the periphery of this bearing is a circular section, rubber ring seated in a groove round the hole in the valve housing and retained by the flange. Another rubber ring, also of circular section, is carried in a groove round the bore of the bearing to prevent the escape of oil along the bearing surface and into the spring housing. The centre portion of the valve has two grooves machined round it, and there are three grooves in the bore of its housing. Two flexible hoses are connected to the valve housing, one is the high pressure supply from the pump and the other is the return to the reservoir. The high pressure pipe is connected to the centre groove in the housing and the return pipe to the other two.

When the valve is in the neutral position, the land between the two grooves on the bobbin is in line with the high pressure, or intermediate, groove in the housing. This land is 0.012 in narrower than the high pressure groove in the housing, so there is a 0.006 in clearance between its edges and those of the groove. Similarly, the lands on each side of the high pressure groove in the housing are 0.012 in narrower than the two grooves opposite them in the bobbin, so there is also a 0.006 in clearance between the outer edges of these lands and grooves. Oil from the high pressure groove passes through the clearances into the two grooves in the bobbin and then out through the other clearances to the return grooves in the housing. Thence it passes to the flexible hose and back to the reservoir.

When the valve is displaced to one side of the neutral position, for example, towards the cylinder, as in C of the

valve diagram, Fig. 4, the high pressure groove is placed in communication with the outer of the two grooves in the valve bobbin and the other groove on the valve bobbin is placed in direct communication with the inner of the two return grooves in the housing. At the same time, the other return groove in the housing is blanked off. In these circumstances, high pressure oil flows into the outer groove of the bobbin and thence through a diagonal drilling through the axis of the component to the annular space between the inner end of the bobbin and its bearing. From this groove, passages are drilled to communicate with the external pipe that carries the oil to the other end of the cylinder. Thus, the whole unit tends to move in the direction in which the valve is displaced.

At the same time, the oil returning from the near side of the piston passes into the annular space between the valve bobbin and the bearing at the other end. Thence it goes through a second diagonal hole to the other groove in the bobbin and to the return groove in the housing and back to the reservoir. If the valve is displaced in the opposite direction, the high pressure oil is fed to the bobbin groove which previously carried the return oil, and *vice versa*. Thus, the unit is moved once again in the same direction as the valve.

Since the effective area of the piston face nearest the end of the cylinder that carries the valve unit is larger than that of the face adjacent to the piston rod, some form of compensation is necessary so that the degree of assistance is the same in both directions. This compensation is effected by the reaction pistons at each end of the valve bobbin. These pistons are of different areas. In fact, the area of the reaction piston at the end of the bobbin nearest to the actuating unit is larger than that at the end nearest the valve spring. When the smaller area of the main piston is under pressure, the oil supply to it passes through the annular space adjacent to the larger reaction piston, and the pressure thus applied to the reaction piston assists the driver in moving the valve. On the other hand, when the high pressure oil is applied to the larger area of the main piston, it passes through the annular space adjacent to the reaction piston of smaller area and again assists the driver to move the valve. The difference between the areas of the reaction pistons is such as to compensate for that between the areas of the two faces of the jack piston.

The movement of the valve required to obtain the full pressure in the fluid is 0.025 in, but movement up to 0.050 in is obtainable. Only in exceptional circumstances, such as turning the wheel against a kerb, is the full movement effected. Under normal conditions, a rate of steering of 12 deg/sec angular movement of the road wheels can be obtained. Although this rate of steering is seldom required

for entering a turn, it is desirable in that it provides for rapid straightening up after the turn has been completed.

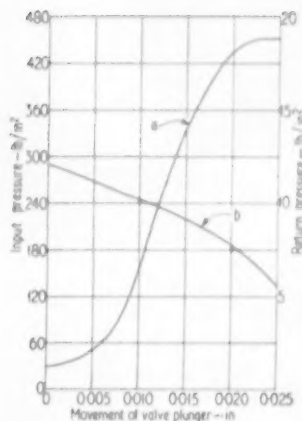
Undoubtedly, the feature that contributes most to the success of this power steering unit is the arrangement of the spring that centralizes the valve. A single spring is employed and it is precompressed to a load of about 75 lb. With most installations, an effort of about 5 lb is required at the steering wheel rim before the spring is compressed further. Thus, until the load on the steering wheel reaches this value, the unit acts in the same manner as a one-piece drag link. As the load on the steering wheel increases, the valve is moved axially and hydraulic power assistance is gradually applied. A typical curve showing the power assistance characteristic is given in Fig. 5. From this it can be seen that the initial rate of increase of hydraulic pressure is very low. In fact, it is not possible to tell, by the feel on the steering wheel, when power assistance begins.

Pre-compression is applied to the spring by screwing the centre component of the three-piece valve assembly on to the end of the outer component that houses the ball joint. In this way, the spring is compressed between a shoulder round the ball joint housing and a flange round the end of the centre component. The diameters of the shoulder and the flange are equal to the diameter of the axis of the wire of the spring. Thus, each only forms half of the seating for the spring; the other half is formed at one end by a shoulder round a hole in the housing into which the flanged centre component fits, and at the other end by the spigot of the housing for the ball joint assembly. This means that the distance between the spigot end and the shoulder must be accurate to relatively fine limits. To obtain the required degree of accuracy, shims are fitted. Each end of the spring bears on a flanged brass sleeve to ensure even distribution of load over the ground ends of the spring when it is compressed. These quickly bed down to accommodate slight differences in the lengths between the halves of the seating faces. From the illustration it can be seen that movement of the valve in either direction further compresses the spring between the housing and the valve assembly.

The other power steering arrangement manufactured by Girling operates on the same principles, but the detail arrangement of the spring and the ball joint assembly attached to the drop arm is different. Square section wire is employed for the spring to obtain the required stiffness in the smallest possible space, and the dimension between the outer seating faces for the spring is controlled by a distance tube. As has already been explained, it is necessary

Fig. 5. A typical curve of the power assistance characteristics

a Input pressure
b Return pressure



to prevent relative rotation between the track rod assembly and the axis of the jack. It is also essential that the location is effected in such a way as not to add to the friction between the valve assembly and its housing, otherwise a stick-slip action might take place and give rise to undesirable steering characteristics. Therefore, the location is effected by two straight rows of ball bearings, diametrically opposite one another and parallel to the axis of the unit. The balls are separated by small coil springs. Their inner races are longitudinal grooves in the outer periphery of the ball joint housing and their outer races are grooves in the inner periphery of a hardened steel sleeve. This sleeve is housed in a cylindrical casting fitted over the end of the valve unit to enclose both the ball joint assembly and the valve spring. The outer end of this casting is closed by a plate, which is retained by a Seeger type circlip in a groove near the end of its bore.

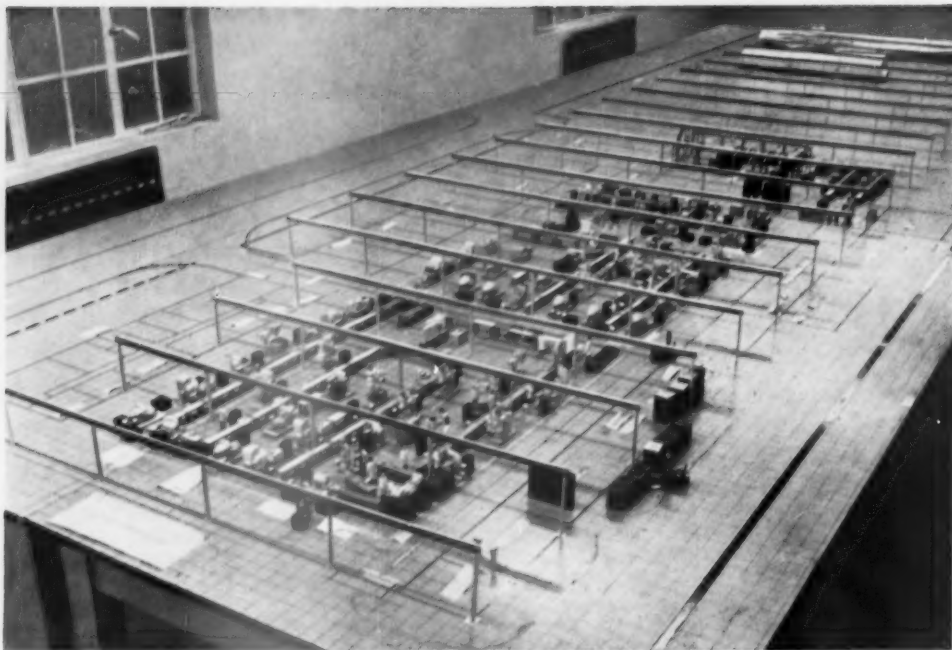
The two cup components of the ball joint attached to the drop arm have spherical seatings machined in their inner faces. So far as their external form is concerned, they are cylindrical in elevation and rectangular in plan. Therefore the ball pin can rotate about its own axis or about a vertical axis through the centre of the ball, but is located against rotation in a vertical plane to prevent relative rotation between the drag link assembly and the hydraulic jack. A saddle piece is fitted against each cylindrical face of the cup components, and the whole assembly is retained by a ring nut in the end of the housing.

Borg-Warner Overdrive

IN the 1955 Annual Show Review issue of the *Automobile Engineer*, in the description of the Borg-Warner overdrive it was stated that if the throttle were held open above the overdrive cut-in speed, the large initial current necessary to operate the solenoid would continue to flow, and also that when the overdrive is locked out, the high current flows through the solenoid as long as the vehicle speed is above the cut-in speed. These statements are incorrect and give a completely wrong impression.

Electrically the solenoid unit contains two separate circuits wired in parallel, (a) the solenoid proper, and (b) an electro-magnet. These circuits are coupled at one end to a lead fed from a relay, while at the other end circuit (b) is permanently earthed, but circuit (a) is broken when the central plunger reaches its operating position. The plunger has an external return spring and carries inside it a spring-loaded pawl rod.

At cut-in speed the large current through the solenoid coil moves the plunger to the right to compress the return spring, and also the pawl rod spring if the pawl is unable to move into engagement. When the plunger contacts the pole piece of the magnet, it breaks the solenoid earth contact, but current continues to flow through the magnet circuit. Since the plunger is then in actual contact with the pole piece of the magnet, a very small current is sufficient to hold it there against the spring load. The position then is that the pawl rod spring is compressed in readiness to move the pawl into engagement as soon as conditions are favourable. This small magnet current continues to flow as long as the vehicle is running at speeds in excess of the cut-in speed, but the large solenoid current flows only for a fraction of a second. The electrical equipment for this over-drive is supplied by Joseph Lucas Ltd.



"Visual Planning" layout of exhaustor production unit in new factory

Vacuum Braking Equipment Manufacture

Methods Introduced at the Clayton Dewandre Works to Cope with Rapidly Expanding Output. PART II.

IN the main machine shop on the ground floor, the addition of plant and equipment over several years in the effort to raise production to keep pace with demand had resulted in a congestion similar to that experienced in the assembly shop. As relatively heavy machine tools and fixed plant were involved the problem called for different treatment, however, and a survey soon revealed that rearrangement of equipment within the confines of the shop could not effect the desired and lasting improvement. As a consequence it was decided to detach one component that, for production, assembly and testing, could be handled as an independent unit in another factory. By this means floor space could be made available in the main shop to permit a worthwhile re-planning to meet existing and future demands for the other products. The component selected for this purpose was the rotary exhaustor, which is produced in eight different sizes and in a range of 12 models, including base-, flange- and spigot-mounted types.

The company is in possession of a newly-built factory at Boultham, a few miles out of Lincoln, and a month ago the necessary plant was transferred from the main shop. This operation was planned down to the last detail so that production should suffer the least possible interruption. Foundations were prepared, shop supply lines were made ready for connecting up, and wiring was run in readiness for the machines and tools which were moved into position at the week-ends.

For all preliminary investigations and for the final layout "Visual Planning" equipment was utilized. An illustration shows the model layout of the production, inspection, palletized store, assembly, testing, and despatch departments

for the exhaustor unit in the new factory. A second, and larger, model layout for the main shop in the parent factory occupies another room. A common policy, sufficiently flexible to permit of minor modification and adaptation to suit the individual requirements of a variety of products, is applied in both the factories. The self-contained layout for the production of the exhaustors may conveniently be taken as indicative of the system in general.

On the production lines of the three main sections—for exhaustor bodies, end covers, and rotors—machines are arranged on both sides of continuously running conveyors of the slat type manufactured by G. W. King Ltd. of Hitchin. Where machines are grouped for sequential machining operations, work may be transferred on gravity roller tracks. Bodies are transferred directly and end covers are palletized but rotors are moved in wooden fixtures in which they are set up individually to prevent the possibility of accidental damage. The conveyors feed to the inspection tables and after the parts have been checked they are loaded directly into pallets by the inspectors. Coventry Climax 1-ton and 2-ton forklift trucks are used to transfer the inspected parts to the palletized store, where matched sets of parts are collected ready for issue to the assembly department. Here the parts are washed, drilled for dowels and passed to the lines.

The assembled components are conveyed direct to the floor-level track in the test department. From this they are picked off and transferred to hangers on an overhead conveyor line serving the rows of test stands. The testers lift the exhaustors from this line to the test stations and,

after the test is completed, re-hang them on the line. The return loop of the conveyor is located immediately alongside the store and the tested units are lifted off and, after a wooden plate has been fitted over the base orifice to prevent the leakage of oil from the interior, are loaded into pallets for transport by road to the engine builders. Fifteen units are carried in each pallet, which is of steel since wooden pallets tend to deteriorate as a consequence of oil leakage from the components.

Production methods

In view of the wide range of the company's products, the multiplicity of individual items, and the varied level of demand, many parts are necessarily batch-produced on general-purpose machines. Whenever practicable, however, specific items or complete components are put into continuous production on a definite time cycle. Rotary exhausters may be cited as a case in point. To detail the operations on certain of the major items of these components is to indicate generally the methods applied in the reorganization of production.

Every precaution is taken to avoid an involuntary interruption of production. Most machines served by powered transfer lines are operated on a strictly scheduled tooling system. After a specified number of hours in operation, determined from a study of the experience in actual working conditions, all tools are withdrawn for servicing. They are immediately replaced by duplicate tools which have been reconditioned in a special tool-servicing shop. The actual changeover is, of course, carried out at the end of a shift. These duplicates are stored alongside their respective machines against the contingency of an individual breakage between scheduled replacements. In the event of such a failure the faulty tool is immediately reconditioned, or is scrapped and replaced, and then restored to its position by the machine.

Lubrication of machine tools and equipment is also arranged as a scheduled service in off-shift hours. A trolley carrying all the required lubricants is brought to the line and a trained operator makes all the necessary replenishments. The line chargehand gives a check signature for



On this Precimax fine borer, exhauster bodies are rough faced and bored and subsequently finish faced and bored

the service, which includes the complete change of spindle oil in the case of fine boring machines.

Machining exhauster bodies

The cast iron body of the exhauster is in continuous production on a line arranged for a four-minute time cycle and worked on a double shift basis. The first operation is to face one end of the casting in order to provide the datum face for subsequent location when boring. At present this is done on a Herbert No. 4 turret lathe modified for use as a facing lathe and equipped with a Wimet tool and a hand-operated chuck with a set of hard jaws.

A transversely arranged gravity type roller track conveys the faced castings to the Precimax duplex, double-ended, fine boring machine. On this they are set up in special

The commencement of the body machining line. On the right is the Precimax fine borer



Fully machined exhauster bodies are passed from the main conveyor on to a roller track for finishing operations



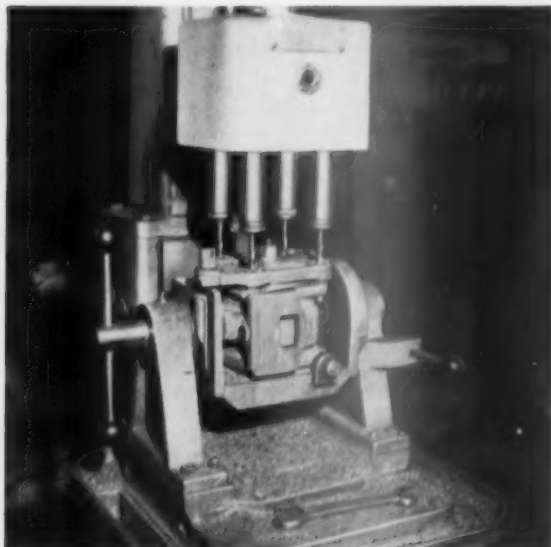
Precimax fixtures on the right-hand heads for the rough facing of the other end of the casting and for rough boring. Dust extraction equipment is provided for this operation.

The castings are then transferred on another transverse roller track back to the other side of the line for two drilling operations. On a Pollard machine fitted with a 4-spindle head, four tapping holes for the cover screws are drilled in each end face. The work is supported in an indexing, turn-over fixture bolted to the machine table. Alongside the Pollard drill is a Herbert single-spindle drill and on this the eight previously drilled holes are countersunk.

After drilling, the bodies are returned on another roller track to the Precimax borer, where they are finish faced and finish bored on the left-hand heads. The operator of this machine unloads and reloads one pair of heads while work is being performed on the other pair of heads. Earlier, the practice had been for the operator to inspect and check dimensions with a plug gauge and a length gauge. In the new layout, Solex air gauging equipment is provided and with this a patrolling inspector will check dimensions of the



Hydraulically operated indexing fixture for milling of the body base and top facings



Turn-over jig on Pollard drill with multi-spindle head for drilling body end faces

bore, the length, and the squareness of bore and faces, and also examine the casting for soundness and freedom from porosity.

Next, the bodies are transferred to a Cincinnati No. 2 vertical miller, where the base and also the top face are machined. A new hydraulically operated indexing fixture has been fitted and on this the work is located from the bore on a substantial overhung spigot. Using a standard 4 in HSS face mill this operation exceeded the scheduled time cycle, so a change was made to a Galtone carbide-tipped face mill. With this tool fitted the operating time was reduced by more than 55 per cent.

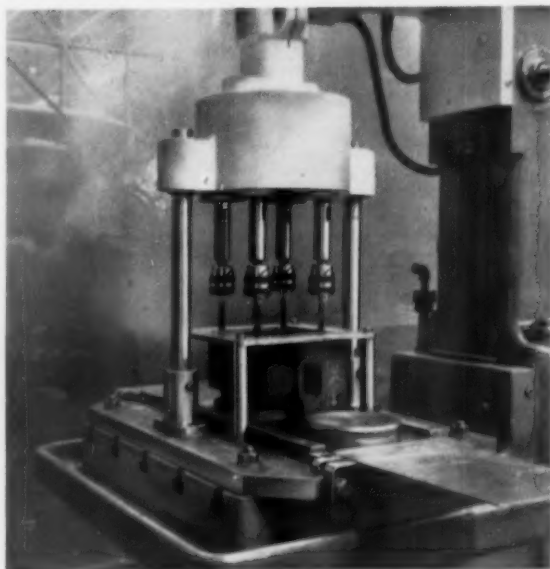
Then, as the body is transferred on the powered slat-type conveyor, follows a series of drilling, bottoming, reaming and spot-facing operations on Kitchen and Wade radial drills. On these machines conventional types of jigs and drill plates are used and the drill spindles are fitted with quick-change collets.

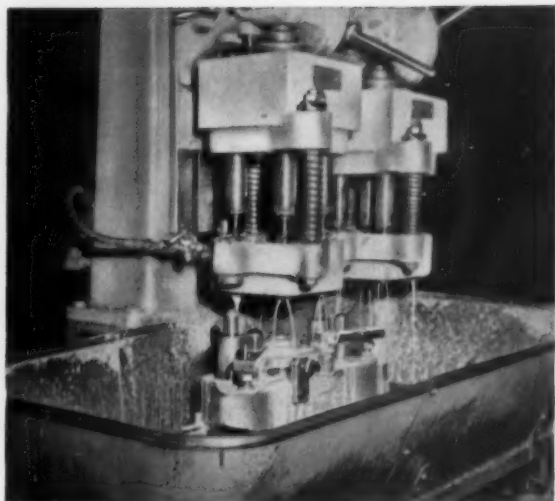
Two slots are then milled under the base for clearing two of the base holes. This operation is done on a Cincinnati

No. 2 vertical miller with a 4 in diameter cutter, $\frac{7}{8}$ in wide. Two tapping operations on Huller automatic cycle machines complete the machining. On the first of these, a single-spindle machine, two holes are tapped $\frac{1}{2}$ in diameter gas thread. As these holes are at different levels, the two-way fixture provided to facilitate loading and unloading is arranged to restore height to suit a single tap setting. The second Huller machine has a 4-spindle head and is used to tap the cover screw holes in the body end faces. This machine also is equipped with a slidable, two-way, loading fixture.

Thereafter the body is conveyed to a fettling station where burrs and sharp edges are removed by wire-brushing. It is then washed in a Laycock paraffin-air spray. A Dawson washer is to be installed later. After final inspection it is

Huller automatic-cycle tapping machine with 4-spindle head. Two-way loading fixture for tapping cover screw holes





Drilling and reaming four holes in end covers on Pollard duplex drill

loaded into pallets ready for transport by forklift truck to the palletized store.

End covers

B.S.A. 9 in automatic chucking machines are used to machine the end covers, which are either of cast iron or of light alloy for different models of the exhaustor. The tool set-up is arranged to face the cover and bore to three diameters, in roughing and finishing cuts. At the first operation all three bores are roughed out and a cross-slide tool rough machines the face. Sequentially, as the tool turret is indexed, the seal housing is finish bored and the ball-race housing is second bored; the bores are chamfered and the

second cross-slide tool finish machines the face; and, finally, the ball-race housing is finish bored to a limit of 0.0005 in.

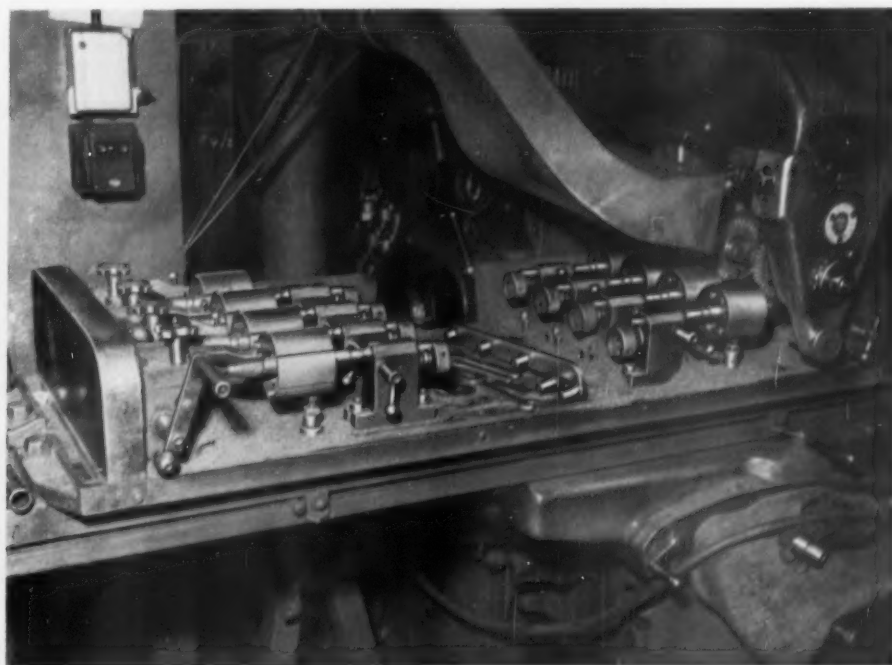
The four holes for the attachment screws are drilled and reamed on a Pollard duplex drill equipped with 4-spindle heads. A slidable work fixture accommodating two covers is indexed below the drilling and reaming heads, and one location is unloaded and re-loaded in each extreme position.

Exhaustor Rotors

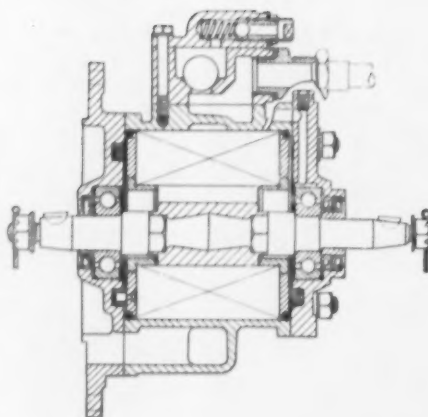
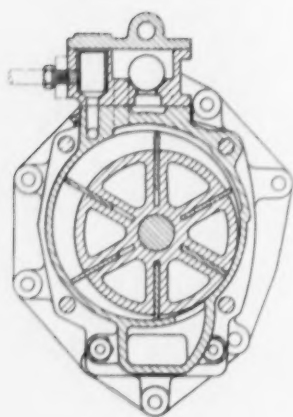
Rotor and shaft are received as a single unit, the solid cast iron rotor being cast on a steel shaft which is suitably profiled to axially locate and to key it in position. The unit is roughed overall in two settings on four B.S.A. 9 in automatic chucking machines. On one of these machines the end of the rotor is faced by a cross-slide tool; approximately half of the outside diameter of the rotor and the three-diameter shaft end are turned; the end of the rotor is recessed; the second cross-slide tool parts off the end of the shaft; the abutment is faced; and the end of the shaft is centre-drilled. It is then transferred to another machine where it is chucked on the machined portion of the rotor and the operations are repeated on the other half of the unit.

The mass of the rotor is reduced by six $\frac{7}{8}$ in diameter through holes running, parallel to the axis, between the blade slots. These holes are drilled on a Pollard drill with a 3-spindle head. The rotor is mounted vertically in an indexing fixture and B.S.A. Penetrator drills are used to drill three holes. The fixture is then indexed through 60 deg and the remaining holes are drilled.

A Cincinnati horizontal miller is employed for gang-milling the blade slots, four rotors being machined simultaneously. For this operation an interesting duplex, indexing work-holding fixture is provided. At each end of the fixture four rotors are mounted between centres, in parallel, and aligned with the four 5 in diameter $\times \frac{1}{2}$ in milling cutters. The rotors are radially positioned by a loose setting jig, shown lying on the fixture in the illustration, which is placed over the rotor shafts adjacent to the back centres. This bar



Duplex, indexing fixture on Cincinnati miller for gang-milling blade slots in the rotors



Transverse and longitudinal sections of rotary exhaustor, flange-mounted type

is furnished with spigots which are entered in the rotor lightening holes to ensure that the blade slots are accurately located in relation to those holes. Then the other end of the rotor shafts are individually clamped to the spindles of the geared indexing head. In the usual manner, one multi-spindle station is unloaded and reloaded while units in the other station are being machined. This machine is provided with dust extraction equipment.

After slotting, the rotor is finish turned on its outside diameter to within a 0.002 in limit. This operation is performed between centres on a Climax lathe, using a Wimet tool. At present the next operation, grinding to finish sizes the three diameters of each end of the shaft, is also carried out between centres on a Churchill external grinder fitted with a 20 in diameter \times 2½ in wheel. Shortly, however, this machine will be taken out, and the operational time will be reduced on a centreless grinding machine, delivery of which is awaited. Solex air gauging equipment is used for checking the shaft diameters, one of which is to receive the inner race

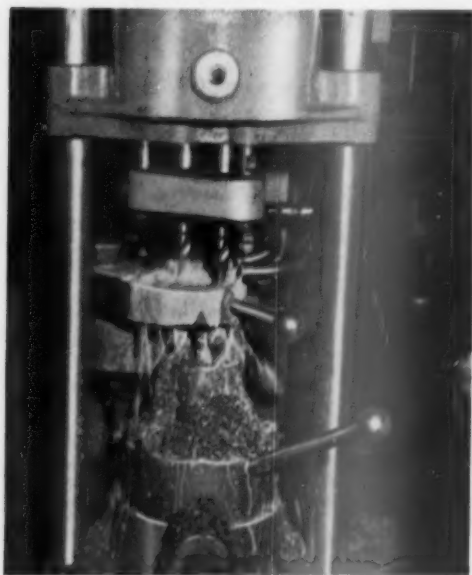
of the ball bearing and is necessarily held to very close limits.

Key slots, for two No. 6 Woodruff keys, are milled simultaneously in the shaft ends on a Cincinnati 1-18 vertical miller equipped with a special duplex-spindle head. Rotors are clamped in two fixtures bolted to the machine table, being set up on vee supports and axially and radially located. As on other layouts, one fixture is serviced while the other is in the operative location. The last operation is to mill two cutter flats on the shaft ends, on an Edgwick single-spindle vertical miller.

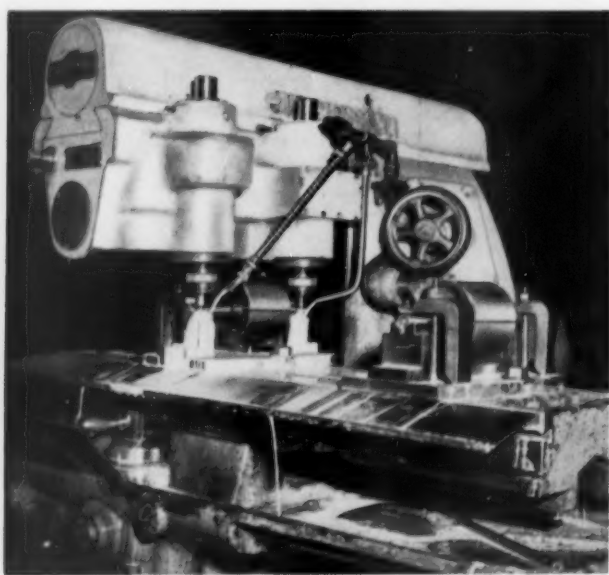
Testing

Every exhaustor is given a full-range performance test before being passed for despatch. On each test bench two laterally mounted units are driven by universally jointed shafts from a single P.I.V. gear. The instrument board mounts a vacuum gauge for each unit and a timing clock. The complete sequence of tests and incidental adjustments occupies about 30 minutes.

Six holes $\frac{7}{8}$ in diameter are drilled through the rotor on this 3-spindle Pollard drill equipped with a 60 deg indexing fixture



Milling Woodruff keyways in shaft ends on special, duplex vertical head Cincinnati miller



The test routine is as follows:

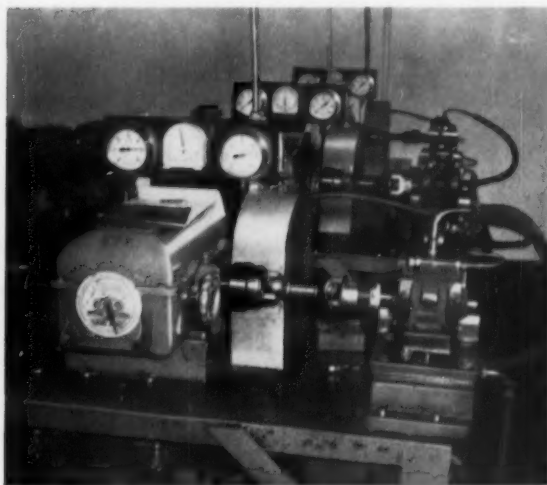
1. Run at 200 r.p.m. and progressively increase to 600 r.p.m. in a period of 3 minutes. During this time the unit is pulling vacuum and oil in order to flood all passages.
2. Decrease speed to 200 r.p.m. and release the vacuum. Close the vacuum valve and start the timing clock. The exhauster must pull 21 in Hg within 44 seconds and

5. Release vacuum, reduce speed to 200 r.p.m., close vacuum valve. Unit must pull 21 in Hg in 52 seconds and the snifting check pressure in 2 minutes.

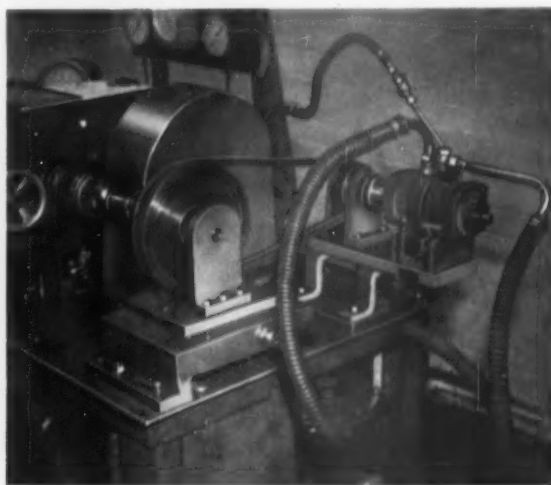
6. Vacuum is released and a run of 10 minutes is made at 200 r.p.m. to drain out the oil.

All test data are recorded on standard sheets.

Usually exhausters are run at half engine speed, but in



Battery of three test benches, each testing two exhausters



Special rig for testing high speed exhausters on standard type test bench

reach the snifting check pressure, up to 26 in Hg according to type, within 2 minutes. The purpose of this test is to prove the performance at low rotational speed, as when a vehicle is coasting with the engine idling.

3. Release vacuum, increase speed to 800 r.p.m., close vacuum valve. The unit must pull 21 in Hg in 14 seconds.
4. With the vacuum valve closed the exhauster is run for 10 minutes at 1,200 r.p.m. to prove the unit is mechanically satisfactory and to raise the operating temperature preparatory to the following test.

some instances units to be operated at engine speed are required. A small diameter, high speed model is produced for such applications and a special set-up is needed to enable these units to be tested on a standard type bench. It comprises a simple increasing drive by rubber belt from a countershaft mounted at the usual test station, as shown in the illustration. Despite the relatively much higher operating temperature resulting from the higher rotational speed, no difficulty is experienced in obtaining and maintaining a performance comparable with that of the more common type of unit operating at half the speed.

Automatic Brake Adjuster

A SELF-ADJUSTING expander unit for hydraulic brakes has been developed by Západočeské Autodružstvo, of Pilsen, Czechoslovakia. Both single- and double-acting units are in production. Each piston of these units has two grooves round its periphery, one for the rubber sealing ring and the other for a spring ring, resembling an engine piston ring. The spring ring is expanded radially to bear firmly against the cylinder bore, and its side clearance in its groove is 0.068 in. It acts as a stop to limit the motion of the piston in the inward direction. When hydraulic pressure is applied, each piston assembly slides outwards, to actuate the brakes, carrying with it the spring ring. Then, when the pressure is released, the piston retracts under the influence of the brake shoe return spring, but its motion is restricted to an amount equal to the side clearance of the ring in the groove, because the friction between the ring and the bore is great enough to resist movement due to tension in the return spring. Thus, the position of the stop is continuously adjusted so that the travel of the piston is constant, regard-

less of wear of the linings or drum. Currently, 2,000 of these patented units are undergoing tests in different types of vehicles.

This self-adjusting expander is made in single- and double-acting units



Decorative Plated Finish

Notes on Electro-deposition Processes

D. J. FISHLOCK

ABY no means uncommon sight in post-war years has been that of sadly deteriorating plating on the exterior of motor vehicles, the condition varying from an incipient corrosion to complete exfoliation, or peeling, of the deposit, and tarnishing or rusting of the article. In this article, some attempt is made to outline the chief troubles, and to give some indication of the solutions available for the apparent decline in the quality of decorative electro-plating.

The applications of an electro-deposited film are many, but may very broadly be divided into three categories:—

- (a) decorative, as illustrated by chromium, silver and gold plating;
- (b) utilitarian, that is, primarily for corrosion resistance, such as zinc, tin and cadmium plates, and
- (c) salvage, the application of thick deposits of nickel, copper, iron or chromium to repair an undersized or damaged article.

These notes are chiefly concerned with the first category, the decorative deposit, although its corrosion resistance is important, since a poorly-resistant film will soon lose its aesthetic appeal.

For the last thirty years by far the most popular plating has been the cold, brilliant, blue-tinged chromium plate. It is this deposit, or more accurately, the nickel-chromium combination which is responsible for most of the controversy relating to the quality of plating. Chromium is a fairly common but interesting element, currently priced at some 7/- per lb, which possesses many peculiar metallurgical properties. For the automobile engineer, the most important one may be summarized as an abnormally high surface activity. It is this feature which causes the metal to react instantaneously with atmospheric oxygen to produce an invisible but impermeable oxide layer a few atoms thick,

which effectively seals off the reactive metal from many adverse environments. It also prevents the adhesion of "soils," as may be seen from attempts to paint a sound deposit, and inhibits "wetting" by liquids, which are summarily displaced unless they react with the oxide film.

Unfortunately, it has a number of features that detract from its usefulness, such as the extreme hardness and low ductility of the plated metal. It was early appreciated, therefore, that it would be impracticable to plate more than a few millionths of an inch of chromium for decorative purposes, since greater thicknesses become increasingly matt, and owing to a hardness of about 1000 V.P.N., extremely difficult to polish. It is hardly surprising, therefore, that such a minute film is by no means continuous. In fact, due to a high internal stress, it is actually permeated by minute cracks and pores, which, although in no way impairing its brilliance, will allow the atmosphere to reach the base metal. Furthermore, because of the peculiarities of bi-metallic corrosion, such a deposit on steel or brass would cause accelerated corrosion at the discontinuities. If, however, another metal which is itself corrosion resistant, although perhaps lacking the brilliance and tarnish resistance of chromium, is interposed between the relatively reactive base metal and the chromium film, a highly durable protective coating will result.

Until the first commercially feasible process was developed for plating chromium—in 1924—nickel was foremost as a low priced, decorative metal finish, although it required maintenance insofar as it tended to fog and develop a yellow tarnish on atmospheric exposure. The quality of the nickel was, in general, very high, being produced manually by highly skilled craftsmen who, though far from conversant with the underlying scientific principles, had acquired a profound knowledge of the practice. It was reasonably logical, therefore, that the first chromium deposition should be made on to a relatively thick deposit of nickel, about 0.001–0.002 in deep. This proved an immediate success, the chromium acting in a manner analogous to a very thin lacquer film. It is interesting that to this day, and despite the advances in the field of the last two decades, no entirely satisfactory alternative combination has yet been evolved, and the nickel-chromium finish is pre-eminent in the realm of inexpensive decorative finishes.

A very brief account has now been given of the composition of a duplex finish, but, of course, many complications are apparent on closer examination. In the first place, electro-deposited metals do not, broadly speaking, plate out fully bright in practicable thicknesses. In the cases of both nickel and chromium, the metals will only deposit fully bright up to some 0.0005 in, and while this depth would be adequate for the chromium layer in the combination, the same depth of nickel would have no protective value whatever on steel, and little on brass. Consequently, after nickel plating, which requires to be some 0.001 in or more thick on ferrous metals and at least 0.0005 in on brass, the deposit must be buffed on a high-speed mop using a very fine, soft abrasive such as lime. Again, it is clearly impractical to deposit a layer of perhaps 0.001 in of nickel, which will follow the contours of the surface, if the latter has imperfections several mils. deep. The work must therefore be prepared to a high finish before decorative plating. The polishing sequence used depends

Polishing hub caps prior to chromium plating at Ford Motor Co. Ltd.



upon the nature of the article and the surface finish, but briefly consists of light grinding operations with successively finer grades of emery, followed by mopping with grease-bonded abrasive compositions.

Electro-deposition of a metal arises from the movement of ions towards, and discharge at, the negatively-charged work in the plating solution, under the influence of a low voltage direct current. Normally the metal being deposited comes from the positively-charged anode. The rate of discharge is directly proportional to the amount of current flowing, and since this will vary in density over an irregularly-shaped cathode partly owing to variations in anode-cathode distance, and partly owing to solution characteristics, the deposit will vary in depth over the cathode surface. For example, holes and recesses will be depleted of metal, while projections will tend to build up. The surface texture of the cathode is also important, since prominences will tend to be exaggerated at the expense of depressions. Another feature of electro-deposits in general, and nickel plate in particular, is that the deposit is pervaded by tiny pores which for a given set of plating conditions demand a certain minimum thickness to prevent their penetration to the underlying metal. Otherwise they can form the sites of a very intense localized corrosion, particularly with iron or aluminium alloys. The adhesion and ductility of the plate, which depend upon the preparatory stages, and on the type of solution and operating conditions respectively, also require close attention.

The normal surface finishes encountered by the plater are rolled, machined, cast and forged, all of which require some degree of mechanical surface preparation, although this need be surprisingly little in the case of well-rolled sheet metal or die-castings. Since the final stages in the polishing sequence invariably involve the application of a grease-bonded abrasive "soap," the polished surface can be expected to retain traces of grease, while holes and recesses may well contain accumulations of grease, abrasive and metal dust. Finger prints, some tarnish, and dust films can also be expected. All these contaminants must be scrupulously removed before plating is attempted, since only an intimate metallic contact in which the inter-atomic forces can participate will give a soundly adhering plate. A pre-treatment schedule, designed to eliminate any and all forms of contaminant which might reasonably be anticipated is therefore necessary. This will have to be a multi-stage sequence including perhaps three or four different types of cleaner, selected according to the shape and composition of the article, and the types of contaminant. Acid treatments will also be necessary to eliminate traces of tarnish or corrosion, while each of the stages will be succeeded by a thorough water rinse.

Since each of the stages plays a specific and vital part in the pre-treatment schedule, it is virtually essential, particularly in the case of mass-production lines, where rejects may be produced in very large quantities in the event of some irregularity occurring, that each stage be amenable and subject to a rigorous control for maximum efficiency. Assuming this very thorough cleaning has been achieved, a pure metal surface, contaminated only by a water film, should be exposed, and this is ready for connection to the L.T. power supply and immersion in the plating bath. In the case of a nickel-chromium deposit an initial, usually very thin, layer of copper may be deposited, after which the work is swilled and transferred to the nickel solution. This is an aqueous solution of nickel salts, with a number of additional chemicals that modify the characteristics of the deposit and facilitate control. It is usually worked warm, up to 50 deg C, and agitated by some means.

The duration of the plating operation depends upon the type and temperature of the solution used, which decide the amount of current which can be passed per unit surface area.



Inspecting finished hub caps at Ford Motor Co. Ltd.

In the latest types of nickel bath, operated at 50 deg C and upwards, the current densities may approach 100 amps per square foot of work surface, yielding an adequate deposit on steel in fifteen minutes. By contrast the old type of cold, static solution required plating times of several hours. The work is then rinsed and either dried preparatory to buffing, or transferred directly to the chromium solution, wherein it remains for some five to ten minutes. A final rinse and dry off finishes the process.

Although the latest types of nickel plating solutions represent a considerable advance, they can also be held responsible for much of the inferior plating of today. It is reiterated that electro-plating solutions do not normally produce bright plates, but there are now quite a range of exceptions, termed bright plating solutions, and usually proprietary in origin. While a theoretical consideration would not be within the scope of this article, and indeed the phenomenon is by no means fully explicable as yet, it has been repeatedly ascertained that the addition of certain elements and compounds, frequently of an organic nature, in very small and closely defined quantities to plating solutions produces very marked changes in the deposit's structure. For example, it has been noticed that the addition of gelatine or saccharin, to mention but two out of thousands of substances, to a dull nickel solution, results in a very bright, hard and brittle deposit. Pains-taking research, directed chiefly at bright nickel developments, has now evolved some particularly striking results in this field, combinations of the addition agents endowing the deposit with a brilliance equal to or bettering the finest buffed plate, while surface imperfections tend to become eradicated with increasing thickness.

The economic importance of such a process is clearly apparent, the intermediate drying and polishing stages being obviated, making the process amenable to full mechanization, while the initial finish on the base metal need not attain such a high standard. The loss of nickel during the polishing process, which may range from 20 to 50 per cent, is also eliminated, while this type of solution is invariably worked faster, thereby further decreasing the production costs.

In view of the advantages of such a process, it is inevitable that there are drawbacks. The chief of these is that most addition agents impart an increased internal stress, which manifests as a tendency for the plate to strip itself from the

base metal and coil up. Consequently an even more rigorous cleaning schedule must be specified, a point which is not always appreciated. In effect, the base metal requires a very slight etch which will "key" the deposit. This factor is a major contributor to faulty plating, particularly that carried out in automatic plants, where a reduction in efficiency of one stage may, if insufficient compensation is available, cause havoc with the reject incidence. The deposit is also very much harder and less ductile than dull deposits, and may develop hair-line cracks if the adhesion is good, in order to relieve the stress. Again, the solution requires considerably more control and maintenance, while the gradual decomposition of addition agents can cause complete failure of the solution or, perhaps worse, a marked reduction in corrosion resistance of the deposit.

The nickel-chromium finish is in popular demand at the present time, both by the customer and because it is now a well established production process. For these reasons a great deal of research is being expended in improving the

finish and solving the outstanding problems. For example, a new chromium plating process has recently been developed which yields a crack-free deposit at very high speeds. There is, however, a world shortage of nickel at present, one which has prevailed for several years and shows no signs of abating. Owing to the importance of the metal in the production of alloy steels and so forth, it is an unfortunate fact that the heaviest burden of restrictions in its use has been borne by the plating industry, which uses some 10 to 15 per cent of the total world output. When supplies are short, there are a number of alternative methods of maintaining production. The most obvious is to reduce the thickness, a method frequently practised, although attended by obvious hazards, and one to which much faulty plating may be attributed. Again, the nickel may be partially or wholly replaced by another metal or alloy, and a number of proprietary developments have been subjected to very searching production and exposure trials. Unfortunately, they have not always fully justified the optimistic claims of their sponsors.

COMMERCIAL VEHICLE TRANSMISSIONS

IN Germany, the need to achieve from a given engine as high an average speed as possible allied to the lowest possible fuel consumption under the most varied traffic and road conditions, the lower unit output favoured by trailer operation, and the greatly increased use of diesel engines have combined to cause a trend towards a much wider reduction range with closely spaced gear ratios. With the introduction of heavy-duty tyres, vehicles of 12-16 tons useful load became popular. Overall reduction ranges were raised to 1:10. Transmissions having six forward speeds replaced five-speed transmissions, though in medium-size vehicles not usually operated with a trailer, five speeds are still considered adequate. In transmissions of several steps, much of the driving time is in the upper gears, and hence at least the first two gears below direct drive must be adapted to more sustained operation.

More gear steps entail more frequent gear-changing, and to lighten the driver's task, gear-changing must be made easier. Though sliding-gear transmissions are still used, most German commercial vehicles are equipped with reduction-gearing transmissions in which dog-clutch changing is provided for all change gears.

To meet severe service requirements, an additional high-speed gear or overdrive may be incorporated in a transmission, often in conjunction with pre-selection which also facilitates gear-changing. Multi-step transmissions consisting of a main transmission and an auxiliary transmission have also been produced. Various systems are employed to facilitate gear changing in such multi-step transmissions. Correct changing is ensured by arrangements permitting changing under load, engaging being achieved by means of friction couplings or brake bands.

Transmissions of the more complicated types in which the driver is completely or partially relieved of the task of gear changing cost far more than manually operated transmissions, and this prevents their widespread use. However, other factors, such as savings in maintenance, possible savings in labour through the reduction of driver strain, and greater driving safety may combine to justify the change-over to a modern-type transmission. Tests of the efficiencies of various types have shown the advantages of torque conversion when it is applied to starting and its use is confined to a limited part of the total speed range. For the main speed ranges, it is and will remain unsuitable, since it is inferior in efficiency to direct transmission and mechanical gears. *M.I.R.A. Abstract 55/12/1.*

TYRE DESIGN

TUBELESS tyres already in use for trucks are of the 15 deg bead drop centre type, in all sizes up to and including replacement for the conventional 11-00×22 size. All major truck manufacturers have adopted tubeless tyres for 1956 models, and trailer manufacturers are expected to follow suit. The advantages of tubeless tyres are listed as simplicity, weight saving, improved mounting, improved safety, protection against punctures and blow-outs, cooler running, improved ride and the elimination of tube and flap. Repairing and retreading will involve some differences in method, but there will be no difficulty. For repairing more serious damage to tubeless tyres, it is hoped that chemical curing section patches will be available in the future.

In tubeless tyres, because of the tight fit of the tyre on the rim, the bead wire, which is of more flexible material, is located in the centre of the structure; load is carried on the base of the bead—the 15 deg. taper. The low flange serves as a stop and centring device. New types of material have replaced the cotton fabric used in the chafers of conventional tyres. The tube has been eliminated, and penetrating objects generally cause only a slow leak, if any leak at all. Since the tyres are not interchangeable with conventional tyres on the same rim, the size markings are different.

For the tyre body, rayon cord has been greatly improved to meet increased speeds and loads. The 1650 denier rayon tyre cord is available in three grades, the super grade having a cord strength of 32-50 lb. Rayon of even greater strength and higher resistance to impact, fatigue and moisture is predicted. Nylon 66 is fast becoming popular as a cord material. A variant known as Nylon 6 may prove of use and may cost less. Steel or wire cord has long been used, particularly on the front wheels of buses, and gives high strength and good tread wear, but has the drawback of high cost. Materials being investigated include Fortisan 36 and Dacron.

Among rubber compounds, a new synthetic with physical properties equal to those of natural rubber has recently been developed and may be satisfactory for large truck and military tyres. It is easier to produce and has greater resistance to tread cracking than GRS. Polyurethane rubber, another new type under investigation, has excellent abrasive resistance. Tread patterns for general highway use and for special requirements are described. Improved "super-abrasion" blacks of finer particle size are being used to obtain better-quality tread compounds. *M.I.R.A. Abstract 55/12/5.*

VEHICLE DEVELOPMENT

Use of Models and Analogues to Save Time and Cost

J. L. KOFFMAN, Dipl.Eng., M.I.Loco.E.

MODELS can be employed to save time and cost in vehicle development. The application of this technique is becoming increasingly desirable, since not only are normal development costs rising constantly, but also the efficiency of current designs is so high that each further advance calls for disproportionately greater effort than the previous one. The technique of using models is highly developed so far as the determination of air resistance is concerned, and before the war, electrically driven models of vehicles were used to investigate the characteristics of front, rear or four-wheel drives, and brakes. However, so far, the use of models or analogues is not as widespread as it merits. Apart from direct economies, additional benefits can be derived from the fact that this technique is an effective means of educating young engineers in the art of rational development.

Cooling systems

Cooling systems are becoming increasingly difficult to design and develop because of modern styling trends, which tend to reduce the frontal area available for the installation of the radiator and also because of the relatively high powered engines now employed. With low drag bodies, the power absorbed by the cooling system can be as high as 15 to 20 per cent of the total available. It includes the power absorbed by the fan and the increase of vehicle drag coefficient C_d due to the adverse effect of cooling air flow. Tests on vehicle models¹ have shown that C_d is increased by 8 to 10 per cent as soon as the air begins to circulate through the cooling system. Apart from this, the fan in some instances absorbs as much as 10 per cent of the power required to drive the vehicle; careful proportioning of the system and disposition of components can reduce this to 2-5 per cent. Models can be used also in the study of the shape of cooling system air passages, and of the effect on the flow past the vehicle of inlet and outlet louvre position.

It is popularly supposed that to interpret the results obtained with models, the data has only to be multiplied by the geometrical scale. However, this is not necessarily so. Dimensional considerations show that the scales of time, mass and force also must be taken into consideration. In other words, there must also be a certain dynamic relationship between the velocities for the model and for the corresponding points in the prototype. The velocity through the model must be of such a magnitude as to ensure that the velocity pressures are in the same ratio as the linear dimensions. For example, if the energy gradient of an air flow under a 3 ft high bonnet, with a velocity pressure of 4 in W.G., is plotted on the side of the bonnet and the whole set-up photographed to illustrate it at 1/12 scale, the bonnet will be shown as being 3 in high and the velocity pressure 0.33 in W.G. A 1 in 12 model should be exactly like the photograph; that is, the velocity pressure should be 0.33 in and the air velocity in the model would equal the velocity in the prototype divided by $\sqrt{12}$.

The results of model tests can be applied to the prototype by the use of non-dimensional factors such as the drag coefficient:

$$C_d = \frac{D}{\left[\left(\frac{\rho}{2}\right)v^2 A\right]} \quad (1)$$

where D lb/ft² is the drag or resistance, ρ lb-sec²/ft⁴ the air

density, v ft/sec the air velocity and A ft² the area. For an area of 1 ft², and with the air at N.T.P., $\gamma = \rho \times g = 0.0763$ lb/ft³, (1) is simplified to:

$$C_d = \frac{\Delta p}{q} = \frac{\Delta p}{\left(\frac{v}{66.7}\right)^2} \quad (2)$$

where Δp is the static pressure and q the velocity pressure, both in inches W.G. However, these equations do not take into account the effect of temperature rise on the volume of air passing through the radiator and the resultant increase of C_d . The drag coefficient of a hot radiator is given with sufficient accuracy by:

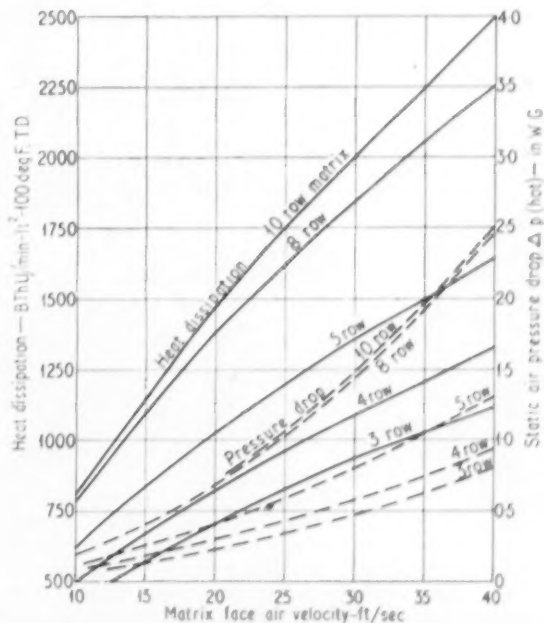
$$C_d = \frac{2T_1/(T_1 + T_2)}{\Delta p_r / (\rho/2)v_1^2} \quad (3)$$

where T is the temperature in degrees absolute, the suffixes 1 and 2 refer to conditions in front of and behind the radiator, respectively, and Δp_r is the pressure loss through the radiator.

From (1), it can be seen that if ρ is constant and D is proportional to v^2 , C_d will be constant irrespective of the velocity. Also, since the values of A , D and v should ensure similarity, C_d of the model should equal C_d of the prototype, were it not for the fact that values of $\Delta p_r/q$ increase as the velocity is reduced, Fig. 2. This means that to obtain similar conditions, the type of flow, laminar or turbulent, should be exactly the same with the model as with the prototype. This requirement can be met by maintaining identical values of Reynolds number, Re , defined as:

$$Re = \frac{vLy}{\mu}$$

Fig. 1. If these radiator performance curves are replotted on a logarithmic scale, it can be seen that the best dissipation rate is proportional to $v^{0.745}$, and the pressure loss is proportional to $v^{1.84}$.



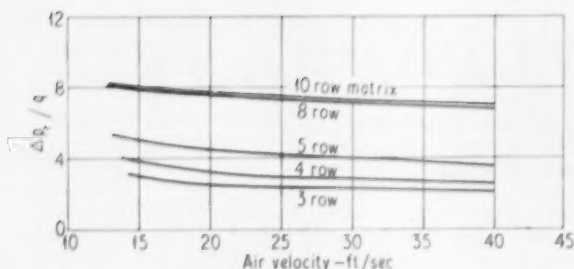


Fig. 2. Curves, derived from Fig. 1, showing how $\Delta p/q$ increases as the velocity is reduced

where v ft/min is the air velocity, L ft a generalized dimension, γ lb/ft³ the density of the air and μ lb/ft-hr its viscosity. The mean value of Re for hot radiators is given by:

$$Re_m = \frac{4r_h v \gamma / \mu_1}{2\mu_1 / (\mu_1 + \mu_2)}$$

where r_h ft is the hydraulic radius, that is, area/wetted perimeter, of each air passage. For Re values of up to 2,350 the flow is laminar so Δp_r , and with it C_d , are proportional to v . At higher values the flow pattern begins to change, and at Re of about 3,500 it becomes turbulent and C_d is proportional to v^2 .

A replot of the radiator performance curves of Fig. 1 on a logarithmic scale shows that the heat dissipation rate is proportional to $v^{0.748}$, whilst the pressure loss is proportional to $v^{1.44}$, that is, turbulent flow is maintained over the normal working range despite the fact that plotting $\Delta p_r/q$ against Re , Fig. 3, suggests that the flow when $v < 20$ ft/sec is laminar. For these curves, the values of Re_m were determined for $t_1 = 60$ deg F and $t_2 = t_1 + (B.Th.U./1.1 \text{ ft/sec})$, since:

$$\Delta t = \frac{(B.Th.U./\text{min-ft}^2)}{c_p \times \gamma \times v \text{ ft/sec} \times 60}$$

where $c_p = 0.24$, and $\gamma = 0.0763$ lb/ft³. The reason for the discrepancy between the experimental and calculated results is probably that the Re values were for smooth passages, whereas the radiator fins are dimpled to promote turbulence. With vehicle models,⁴ the values of C_d vary but little over a wide range of Re values, Fig. 4, although with regular

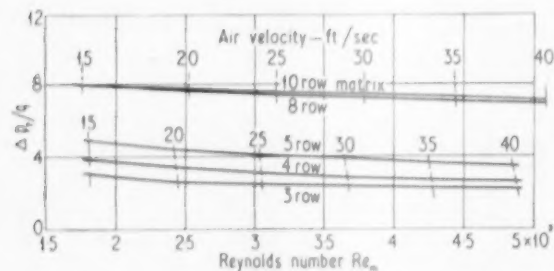


Fig. 3. These curves suggest that laminar flow is maintained when $v < 20$ ft/sec

shapes such as a sphere,⁵ the value of C_d remains virtually constant at 0.4 for Re values of 2,000 to 200,000 and then drops to 0.1 between $Re = 300,000$ and 600,000.

Thus, if model tests are to represent full scale conditions, the Re of the model theoretically must be equal to Re of the prototype. If the scale is $1/12$ and γ and μ are the same for the model as for the full scale vehicle, the velocity through the model should be twelve times that of the prototype. To obtain this velocity, considerable power will be required to drive the blower, so it might be more practicable to employ a larger model. Fortunately, the curves of $\Delta p_r/q$

against Re , Fig. 3, show that a strict adherence to identical Re value requirement is not necessary, since in any case a model can rarely be made identical in every respect with the prototype. For example, the appropriate values of surface roughness are unlikely to be obtained. The investigations should aim at qualitative indications of prototype performance as affected by various design changes. Thus, although the Re values need not be strictly identical, C_d , or $\Delta p_r/q$, should be the same for the model as for the full scale vehicle.

Investigations can be carried out with a scale model of the engine compartment, connected to a source of air supply, Fig. 5. The radiator matrix can be replaced by one or more wire screens, provided care is taken to obtain the appropriate $\Delta p_r/q$ versus Re or V characteristics. The drag coefficient of

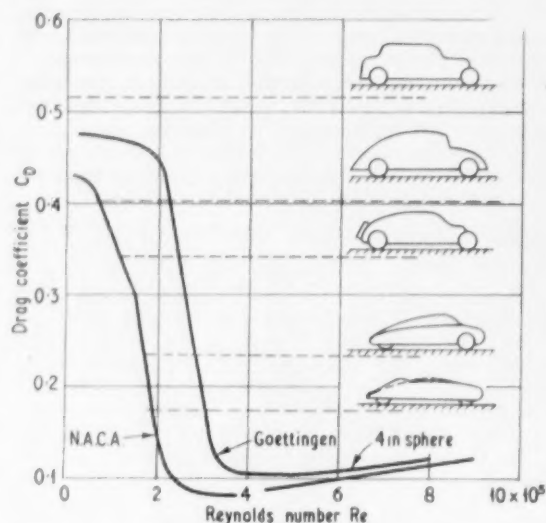


Fig. 4. The drag coefficient C_d for vehicle models varies but little over a wide range of Re values

screens can be obtained from curves.⁶ To begin with, measurements should be taken over a range of velocities and the results plotted in terms of C_d or $\Delta p_r/q$ against Re . This preliminary work is necessary to ensure that the experimental results are obtained within the limits of the horizontal portion of the curve. The data subsequently recorded will serve to indicate the fan and radiator proportions required. Time required for development work carried out on these lines is much less than if the tests had been carried out from the outset on full scale prototypes, and the cost also is much less.

Vehicle vibration models

Bouncing of a vehicle on its suspension springs as it passes over rough roads generally is accompanied also by pitching, since the resultant of the forces on all four wheels rarely acts through the centre of gravity. Pitching about the centre of gravity can be maintained only if the spring stiffness at each wheel is inversely proportional to the distance between the centre of gravity and the effective line of action of the spring reaction. If c_b does not equal c_a , where c is the spring stiffness in lb/in, and a and b the relevant distance in inches, Fig. 6, pitching will lead to bounce; the more nearly alike are the natural frequencies of the two modes of vibration, the closer is the coupling between them. The problem is that if a vibrating system has more than one degree of freedom, one mode of vibration is likely to influence the others, because the vibrations are coupled by the mass.

Bouncing⁴ is defined as a vibration, the centre of oscillation

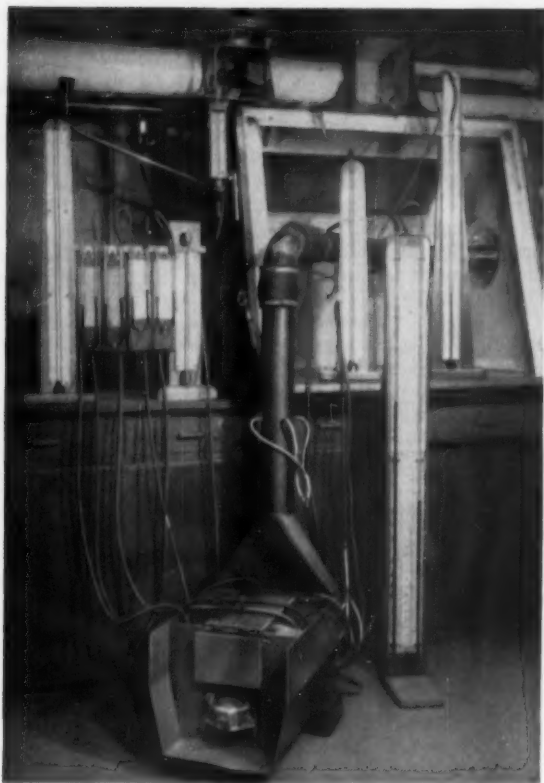


Fig. 5. Apparatus for experimental investigations with a scale model of an engine compartment

of which is outside the wheelbase, and its natural frequency is given by:

$$\omega_b = \sqrt{(c_f + c_r)/m}, \text{ sec}^{-1}$$

or $f = (60/2\pi) \sqrt{(c_f + c_r)/m}, \text{ min}^{-1}$

where c_f and c_r , lb/in, are the stiffnesses of the front and rear springs, respectively, and m , lb-sec²/in, is the sprung mass. The natural frequency of pitching, that is, vibration with its centre of oscillation within the wheelbase, is given as:

$$\omega_p = \sqrt{(c_f a + c_r b)/I}, \text{ sec}^{-1}$$

also, $f = (60/2\pi) \omega_p, \text{ min}^{-1}$

where I lb-in-sec² is the moment of inertia of the spring mass about the lateral axis through the centre of gravity. The coupling factor for bouncing is:

$$k_b = \frac{(c_f a - c_r b)}{(c_f + c_r)} \text{ in}$$

It is geometrically represented by the distance between the axis of resultant spring force R and the centre of gravity, Fig. 6. For pitching:

$$k_p = \frac{(c_f a - c_r b)}{(c_f a^2 + c_r b^2)} \text{ in}^{-1}$$

When ω_b , ω_p , k_b and k_p have been determined, it is possible to calculate the two natural frequencies of the undamped oscillations. These frequencies are given by:

$$\omega_1 = \sqrt{\frac{(1 + \epsilon^2)}{2} - \frac{\sqrt{(1 - \epsilon^2) + k^2 \epsilon^2}}{4}} \quad (4)$$

and

$$\omega_2 = \sqrt{\frac{(1 + \epsilon^2)}{2} + \frac{\sqrt{(1 - \epsilon^2) + k^2 \epsilon^2}}{4}} \quad (5)$$

where $\epsilon = \omega_b/\omega_p$ and $k^2 = k_b \times k_p$. Provided ϵ and k^2 are known, the values of ω_1 and ω_2 can be determined from Fig. 7.

As the result of this coupling effect, the vehicle body

oscillates about two axes. The distance of these axes from the centre of gravity can be derived from:

$$R_1 = \frac{(\omega_1^2 - 1)}{k}$$

and

$$R_2 = \frac{(\omega_2^2 - 1)}{k}$$

Positive values of R_1 and R_2 indicate that the axes of oscillation P_1 and P_2 respectively are behind the centre of gravity and *vice versa*. Furthermore, $R_1 \times R_2 = i^2$, where i is the radius of gyration of the sprung mass.⁴

To illustrate clearly the coupling of the pitching and bouncing modes, a simple model can be constructed, Figs. 8 and 9. The assembly comprises two bars, arranged to pivot, in effect, in a single vertical plane. One bar is approximately horizontal and its right-hand end, as seen in the illustrations, is pivoted to a bracket that is fixed relative to earth. Its other end is pinned to the left-hand end of the second bar, which extends to the right. This bar is offset slightly to one side of the first so that it is free to oscillate in a vertical plane about its pivot. Oscillatory motion is imparted to each bar by a separate motor, driving a crank and connecting rod mechanism. The crank that actuates the first-mentioned bar rotates about an axis that is fixed relative to earth, that which actuates the second bar is on a bracket fixed to the first. At the right-hand end of the mechanism, the pivot on the bracket that is fixed relative to earth is P_1 , and the pivot on the extreme left represents P_2 . The point of attachment of the crank to the first rod is a distance R_1 from P_1 , and the point of attachment of the second crank to the other rod is a distance R_2 from P_2 . The length of the connecting rods does not matter since the operating mechanism is used solely to impart sinusoidal motion to the model. The frequencies of oscillation about P_1 and P_2 are given by:

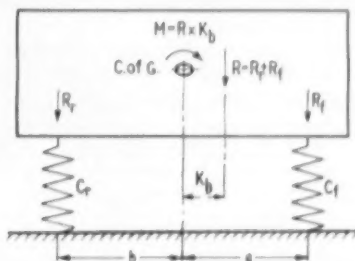


Fig. 6. The coupling factor for bouncing is represented by the distance between the line of action of the resultant spring-force R and the centre of gravity

$$\omega_{P1} = \omega_1 \times \omega_p$$

and

$$\omega_{P2} = \omega_2 \times \omega_p, \text{ sec}^{-1}$$

respectively, where ω_{P1} and ω_{P2} are the frequencies about P_1 and P_2 respectively, ω_1 and ω_2 are the rotational frequencies of the lower and upper cranks respectively, and ω_p is the pitching frequency of the model in Fig. 8. The length of R_1 and R_2 can be adjusted to suit the circumstances, and the frequencies of oscillation can be altered by regulating the speeds of the driving motors.

The following example illustrates the application of the principles outlined. A car has a sprung weight of 2,660 lb, and its centre of gravity is so placed that the distance a , from the front axle, is 61.5 in and the distance b , from the rear axle, is 45 in. The spring stiffness is 475 lb/in at the front, and 175 lb/in at the rear, and $I = 25,600$ lb-in-sec².

$$\begin{aligned} m &= \frac{2,660}{380} = 6.9 \\ \omega_b &= \sqrt{\frac{(475 + 175)}{6.9}} \\ &= 9.3/\text{sec} \end{aligned}$$

$$\omega_p = \sqrt{\frac{(475 \times 61.5^3 + 175 \times 45^3)}{25,600}}$$

$$= 8.77/\text{sec}$$

so,

$$\epsilon = \frac{9.3}{8.77} = 1.06$$

and

$$\epsilon^2 = 1.125$$

Furthermore,

$$k_s = \frac{(475 \times 61.5 - 175 \times 45)}{(475 + 125)}$$

$$= 30.5 \text{ in}$$

and

$$k_b = \frac{(475 \times 61.5 - 175 \times 45)}{(475 \times 61.5^3 + 175 \times 45^3)}$$

$$= 0.00927$$

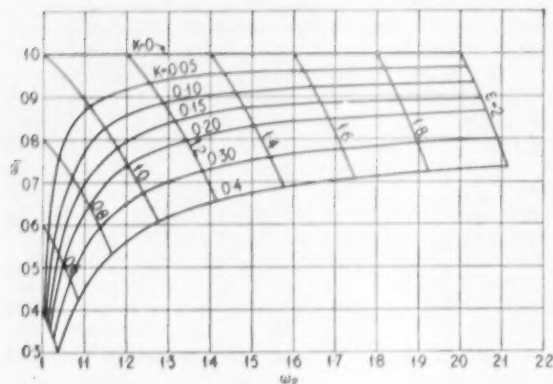


Fig. 7. Curves showing the relationship between ω_1 , ω_2 , k and ϵ

$$\text{so, } h^2 = 30.5 \times 0.00927$$

$$= 0.285$$

From these values, it is possible to determine ω_1 and ω_2 , either from a model of the type illustrated in Fig. 8 or from equations (4) and (5). In this example, $\omega_1^2 = 0.4675$ and $\omega_2^2 = 1.6575$, so:

$$R_1 = \frac{(0.4675 - 1)}{0.00927}$$

$$= -68.2 \text{ in}$$

and

$$R_2 = \frac{(1.6575 - 1)}{0.00927}$$

$$= 70.75 \text{ in}$$

The frequencies of oscillation about P_1 and P_2 are:

$$\omega_{P_1} = 0.684 \times 8.77$$

$$= 6$$

and

$$\omega_{P_2} = 1.285 \times 8.77$$

$$= 11.25$$

so $f_1 = 57 \text{ c/min}$ and $f_2 = 107.5 \text{ c/min}$. The results obtained in practice will be modified slightly, of course, by the suspension dampers and tyres. A model of this type leads to a better appreciation of the pattern of vehicle motion, and its effect on passengers, particularly at head and shoulder level.

Electro-mechanical analogy

When considering vibrations as encountered in mechanical systems on the one hand and in electrical circuits on the other, the assistance obtained by studying the mechanical problems with the aid of an electrical analogue can be readily appreciated. It is essential for the vibration modes in each system to obey identical laws. In some applications it is

desirable to consider a mechanical system in terms of its electrical analogy, since much more is known of the characteristics of electrical circuits than of certain mechanical systems. Also, whilst in classical mechanics, vibrational phenomena are represented by differential equations, electrical engineers can use schematic diagrams to estimate the performance of a circuit without solving its equations.

Whereas in electrical circuits, the voltage is measured by connecting a voltmeter across the two terminals of the element, to measure current it is necessary to break into the circuit. In mechanical devices, velocity or displacement can be measured without disturbing the machine, but force cannot be measured without breaking into the device. Therefore, force is analogous to current, and if a mechanical element is to be analogous to an electrical one it must have a velocity difference between, or across, its two terminals and a force acting through it.

Electrical capacity represents mass, since the energy stored in the electrostatic field of the condenser, $(1/2) CE^2$, corresponds to the kinetic energy of the mass, $(1/2) mv^2$; also, the condenser, by absorbing a current of magnitude CdE/dt , tends to prevent a change of e.m.f. just as a mass tends to prevent a change of velocity, by producing a reaction force mdv/dt . Similarly, inductance is analogous to mechanical stiffness, because the energy stored in the magnetic field of the inductance, $(1/2) LI^2$, corresponds to the energy stored by a spring $(1/2) cF^2$; also, an inductance stores a voltage impulse $\int E dt$, of magnitude LI proportional to the current, while a spring undergoes a displacement, cF , proportional to the applied force.

It is assumed that all springs follow Hooke's law and all mechanical resistances are directly proportional to the velocity between the two points under consideration. In addition,

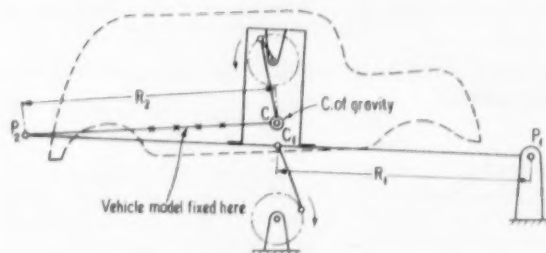
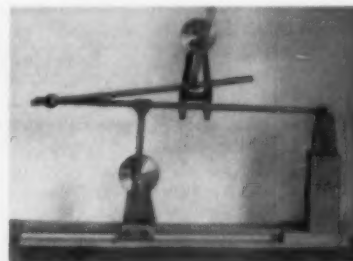


Fig. 8. In the model used to demonstrate the coupling of pitching and bouncing modes, two motors are employed

the forces are assumed to be sinusoidal so that all mechanical elements have simple harmonic motion. If a force is not sinusoidal it must be represented by a Fourier series and the effects combined by superposition.

In this manner, it is possible to establish the following analogy, originated by Firestone,⁶ and generally referred to as the *New Analogy*, the *Mobility Analogy* or the *Electro-magnetic-Mechanical Analogy*:

Fig. 9. The model used to demonstrate the coupling of the pitching and bouncing modes of vibration of a vehicle on its suspension springs



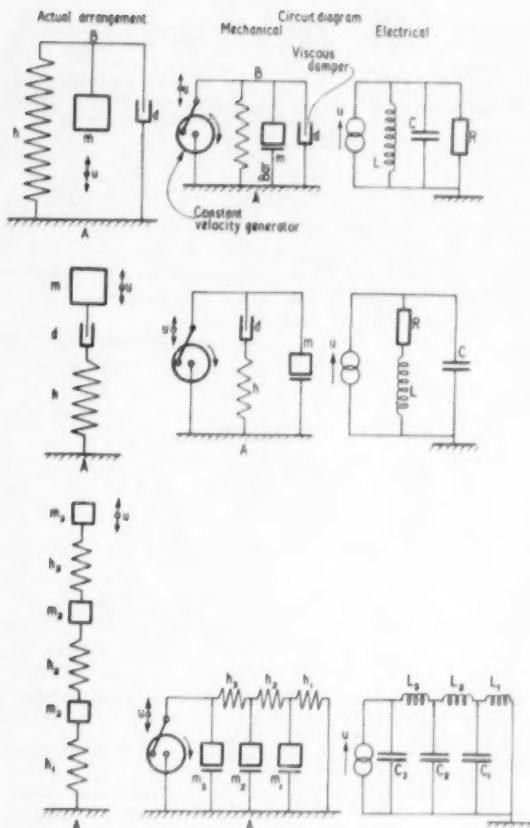


Fig. 10. Method of designing an electrical analogue to represent a mechanical system

Mechanical	
Force through	P lb
Velocity over	v in/sec
Displacement	s in
Impulse	p lb-sec
Inverted spring stiffness	k in/lb
Inverted damping constant	r in/lb-sec
Mass	m in/lb-sec ²
Electrical	
Current through	I Amp
E.M.F. over	E Volt
Voltage impulse	Edt Volt-sec
Charge through	q Coulomb
Self-inductance	L Henry
Resistance	R Ohm
Capacitance	C Farad

The velocity difference at the ends of a mechanical com-

ponent corresponds to the voltage difference at the ends of an electric circuit component, whilst the force passing through a mechanical component corresponds to the current passing the electrical component. Thus, the mechanical circuit can be considered as a mechanical force circuit composed of three basic elements, spring, mass and damper. At first sight, mass appears to be a one-terminal quantity because only one connection is required to set it in motion. However, the force acting on a mass and the resultant acceleration are referred to earth inertia-frame, so that in reality the second terminal of mass is earth. The symbol used to represent mass is shown in the circuit diagrams, Fig. 10. The lower end of the mass moves with a velocity u with respect to the ground, and the bar represents the second terminal of the mass and has zero velocity. To measure the force, a suitable device can be inserted between the square and the next element or generator connected to it. In the diagram, the mass is indicated as related to a point of reference for the acceleration that results from the action of the force.

Since, according to Newton's law, velocity must be defined relative to a system of co-ordinates which move in space without being subjected to acceleration, to meet these requirements one plate of each condenser representing a mass must have a constant potential. This means that the condensers must be connected to earth. The velocity differences at the end of a mechanical element and the potential differences at the ends of an electrical element are similar, as also are the forces passing through a mechanical element and current through an electrical element.

The general procedure of drawing up analogous circuits is indicated in Fig. 10. It is useful to draw first a mechanical circuit diagram, noting the junction points of each pair of elements. This locates all element terminals that move at the same velocity. The circuit drawing is made by attaching to the bottom bar all element terminals that have velocity u , and all terminals with zero velocity to the top, or earth, bar. This circuit is represented by a mobility-type analogous circuit simply by substituting the analogous electrical elements for the mechanical ones.

Thus, in Fig. 10, the mass m is maintained mechanically in a state of motion at sinusoidal velocity of angular frequency ω , the r.m.s. magnitude of which is v ft/sec. The mass is supported relative to a fixed plane of reference, earth, by a spring h and damper d . A diagram is drawn to represent the relevant mechanical circuit in which one end of each of the elements m , h and d is connected to the fixed plane A . Thus, the elements are connected in parallel to each other and to the force P applied between A and B . The analogous electrical circuit is similar, the elements having one terminal connected to earth.

To apply the analogy to a car it is necessary to assume a mass distribution and spring stiffness as shown in Fig. 11. The spring mass is replaced by three masses. Two of them, m_1 and m_2 , are over the front and rear axles respectively, while m^3 is a virtual mass at the centre of gravity. Their

Fig. 13. Layout of the components of the circuit shown diagrammatically in Fig. 12



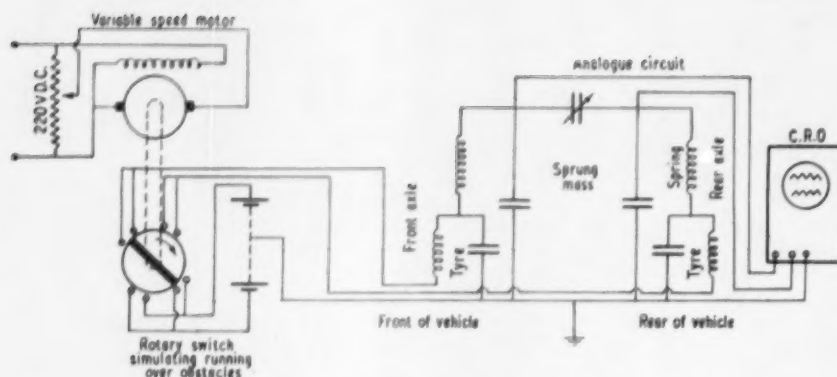


Fig. 12. Left: Electrical analogy for a car with the mass distribution illustrated in Fig. 11

Fig. 11. Representation of the mass distribution in a car

magnitudes must be such as to satisfy the following three requirements:

(a) $m_1 + m_2 + m_3 = m_v$ where m_v = the sprung mass of the vehicle.

(b) The position of the centre of gravity must be identical with that of m_3 ; that is:

$$m_1 a = m_2 b$$

(c) The resultant moment of inertia, $I = m_1 i^2$, about the transverse axis through the centre of gravity must be identical with that of the mass m_3 ; or:

$$m_1 a^2 + m_2 b^2 = m_3 i^2$$

These requirements are met for:

$$m_1 = \frac{m_3 i^2}{a l}$$

$$m_2 = \frac{m_3 i^2}{b l}$$

$$m_3 = m_v \left(\frac{1 - i^2}{ab} \right)$$

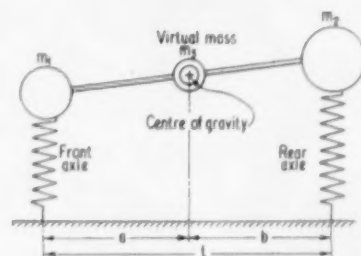
where $l = a + b$.

If the virtual mass m_3 is large, an impact at the front will cause m_1 to move upwards and, because of the inertia of m_3 , m_2 will move downwards. Thus, the body pitches about m_3 . However, if $m_3 = 0$, an impact at m_1 will send only m_1 upwards since in these circumstances m_1 and m_2 can oscillate independently of each other. Again, if m_3 is negative, a displacement of m_1 will cause a similar displacement of m_2 ; that is, the car will bounce, since the centre of rotation will be outside the wheelbase. The mechanical circuit, Fig. 11, can be replaced by the electrical circuit Fig. 12, from which it is possible to study the oscillation of the vehicle body when passing over obstacles. Fig. 13 is a half tone illustration of the circuit shown in Fig. 12.

The author is indebted to Mr. F. G. H. Tutt, B.Sc., for the development of the analogue shown in Fig. 12. Figs. 5, 9 and 13 are Crown Copyright reserved, reproduced by permission of the Controller, H.M. Stationery Office.

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Mechanical Handling Exhibition

THE fifth Mechanical Handling Exhibition will be held at Earls Court, London, from 9th to 19th May, 1956. Since the first exhibition was held in 1948, each successive exhibition has been an improvement on its predecessor, and there is not the slightest doubt that this will hold true of the forthcoming event.

There will be more than 250 exhibitors to make this the largest display of mechanical aids and labour saving equipment for industry that has ever been held throughout the world. Exhibits of actual equipment will number several thousands; they will be reinforced by working models, diagrams, photographs and films, and there will be matters of interest for engineers from any type of industry. In addition, there will be a Convention at which papers on a wide variety of topics will be presented by experts from several countries.

The exhibition is organized by our associated journal, *Mechanical Handling*. Works parties will be particularly welcome and will be given every help and facility. Tickets of admission can be obtained free on application to:—H. A. Collman, *Mechanical Handling*, Dorset House, Stamford Street, London, S.E.1.

Hydraulic Drawbar Dynamometers

*Standardized Instruments to Ascertain or Check the Performance
of Tractor Vehicles*

A TRAILER fitted with pneumatic-tyred wheels has a rolling resistance of approximately 30 lb/ton on a hard-surfaced road and considerably higher on a dirt road. Off the road, but on firm ground, it may well be from 80 lb/ton to 100 lb/ton, according to the character of the surface, under dry weather conditions. After heavy rain or after a thaw the figure may rise sharply to as high as 400 lb/ton.

Whether designing tractor vehicles, quoting for haulage equipment or planning haulage operations, an assessment of the drawbar pull required to haul specific loads over a given route at a stipulated speed and a knowledge of the pull developed at various speeds by available vehicles are essentials. Requirements can be determined and the performance of the tractor vehicle can be evaluated or checked under actual or simulated conditions by a drawbar dynamometer.

Some years ago the Institute of Agricultural Engineering designed and produced a simple hydraulic dynamometer for its own use in checking the performance of tractors hauling cultivating implements, on general farm transport, and in extracting timber. The Institute had not the necessary facilities for the manufacture of the instrument to meet general demand and it was not economic to make one or two at a time to meet special orders. Accordingly, as the demand continued to increase, the Institute in 1949 entered into arrangement with Roadless Traction Ltd., Gunnersbury House, Hounslow, Middlesex, to manufacture and supply

the dynamometers. They are produced to the design of the N.I.A.E. and every instrument is tested and calibrated at the Institute's establishment at Wrest Park, Silsoe, Bedfordshire.

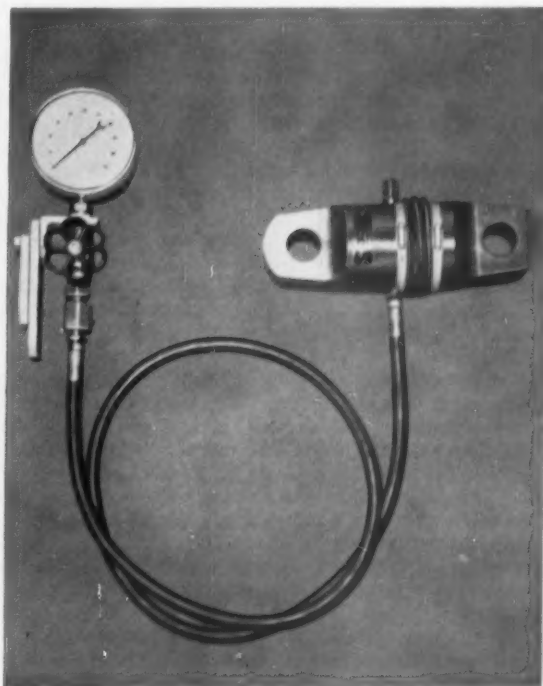
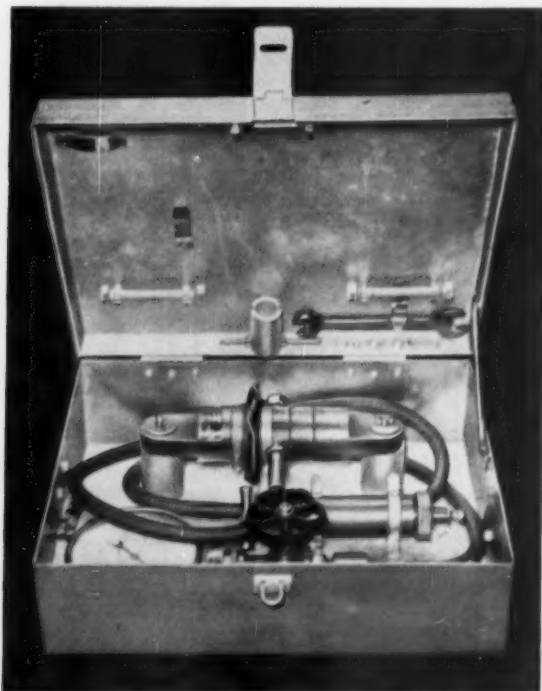
The original dynamometer had a maximum working capacity of 10,000 lb. If desired, it could be calibrated to read to a maximum of 5,000 lb and, if required, two 10,000 lb instruments could be coupled in parallel by a special balanced linkage to give a maximum capacity of 20,000 lb read on a single dial. Later a larger model of similar design was developed having a capacity of 40,000 lb. Two of these instruments may also be coupled together to extend the direct-reading range to 80,000 lb. These capacities cover all general requirements and production of the two instruments is now standardized. Dials are calibrated in pounds, kilogrammes, or in both units.

Obviously, with a dial calibrated to 5,000 lb maximum a somewhat more precise reading can be made of pulls at the lower end of the scale. For all practical purposes, however, the standard 10,000 lb maximum scale is quite satisfactory for pulls down to 400 lb.

It may be queried whether a 10,000 lb instrument to be used for light loadings of, say, half a ton could be calibrated to give a full-scale reading at 1,500 lb or 2,000 lb. While there is no mechanical or hydraulic reason to prevent this, the margin of error under a static load might, as a result of frictional hysteresis, be regarded as undesirably high in

Shown boxed complete with tools and accessory equipment, the smaller dynamometer can be calibrated to indicate a maximum pull of either 5,000 lb or 10,000 lb

The larger model for drawbar pulls up to 40,000 lb. Two of these units can be coupled in parallel to register pulls up to 80,000 lb



Sectional arrangements of 10,000 lb and 40,000 lb dynamometers. Hardened bushes are fitted to the coupling eyes

relation to the average pulls to be measured. In the normal run of trailer haulage, of course, the load is subject to frequent fluctuation and this tends to cancel out the plus or minus effect of the inherent frictional error. However, it is recommended that the instrument should not be calibrated for full-scale readings of less than 5,000 lb.

The design aimed at producing a precision instrument that was sufficiently robust to stand up indefinitely to the rough handling inevitably encountered in field work and also the continual hammering entailed in its secondary function as a drag link between tractor and trailer. So well did the original design developed by the N.I.A.E. fulfil these requirements that only minor modifications to facilitate manufacture have since been introduced.

To-day they are being used in all parts of the world not only for agricultural and haulage tractors but for tests on a wide variety of other mechanical equipment, including winches, capstans, cranes, locomotives and tug-boats. Each dynamometer, complete with 7 ft of flexible hose, gauge, damping valve and adjustable two-bolt attachment bracket, is housed in a stout galvanized steel case for safe storage or transport. The case is equipped with a set of spanners, oil filler gun, spare seals and cup washers, and bolts for attaching the gauge to the tractor.

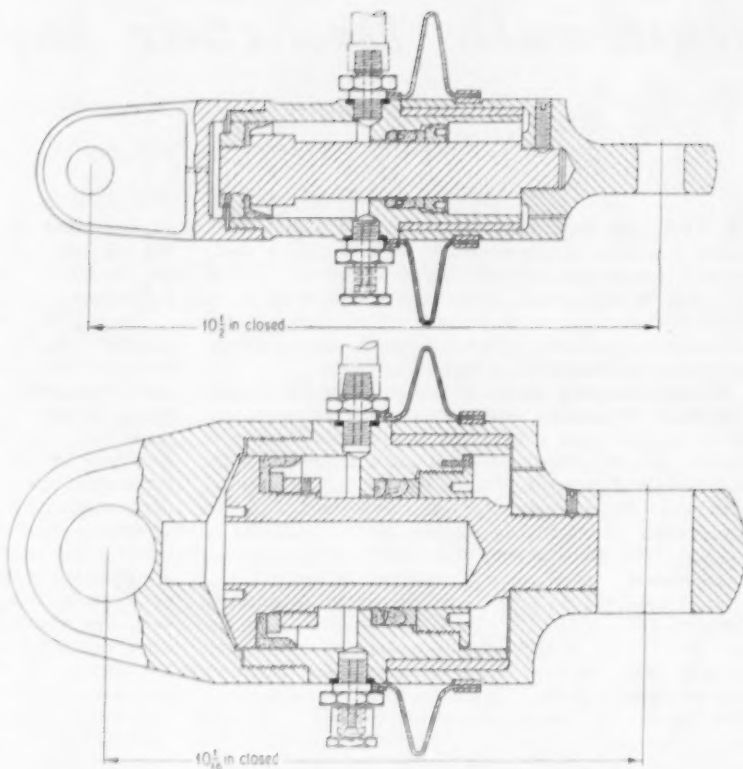
Power absorbed on gradients

With a given tractor vehicle, its suitability for a specified haulage operation can be evaluated if the following data, as a minimum, are known:

1. Total weight of the tractor in working order
2. Drawbar pull of the tractor in its various gears
3. Maximum up-gradient along the route to be taken
4. Weight of the trailer

From these it is possible to estimate what load can be hauled at what speed over the selected route. Reasonable allowance must be made, in the absence of recorded experience, for adverse conditions under which rolling resistance may be increased and drawbar pull reduced as a result of wheel spin or track slip.

For example, a tractor weighing 3 tons and having three gears giving speeds of 6, 11 and 16 m.p.h. has drawbar pulls of 11,000, 6,000 and 4,125 lb respectively. If the maximum gradient to be encountered is 1 in 16, and assuming that lack of adhesion of the tractor vehicle will not arise, it is desired to ascertain if the tractor will haul a net load of 10 tons on a trailer weighing 3 tons and in which gear it can negotiate the gradient. The gross weight of the tractor and the laden trailer is 16 tons and, therefore, on the gradient gravity will absorb one-sixteenth of the gross weight, that is 2,240 lb, in reduction of the drawbar pull available for overcoming the rolling resistance of the trailer. This would leave 8,760 lb in first gear, 3,760 lb in second gear and 1,885 lb in top gear.



Performance estimates

Obviously, on firm ground and under dry conditions, when a rolling resistance of 100 lb/ton or 1,300 lb could be safely assumed, the tractor could handle the load in top gear with a satisfactory margin of power in reserve. Should the track deteriorate with continued use or should adverse weather prevail and the rolling resistance be doubled at 200 lb/ton, the gradient could easily be surmounted in second gear. Even under the worst conditions, with the rolling resistance at 400 lb/ton, the run could be made in second gear and first gear used only for the maximum gradient.

Such estimates of performance should always be of a conservative character in order to avoid trouble arising in operation. It is advisable never to load a tractor so that normally it cannot operate in an intermediate gear; bottom gear should always be in reserve to meet emergencies. If a tractor is engaged on work demanding bottom gear continuously, it must be regarded as overlaid, and sooner or later trouble will be encountered.

Relationship of speed and drawbar pull

It is of convenience to determine the drawbar horse power of a tractor by means of the formula:

$$\frac{\text{Drawbar pull (lb)} \times \text{Speed (m.p.h.)}}{375}$$

375

This figure remains substantially constant and, presuming satisfactory tractor adhesion is maintained, the constituent factors can be relatively adjusted to give the most economical operation over a given route. For instance, the desired load may be somewhat too high for the tractor to operate throughout the run in top gear and, conversely, too low for economic operation in second gear. Either a reduced load could be hauled in top gear or in the second gear a heavier load could be transported. If the gear ratios could be suitably modified,

the desired load could be hauled at a slightly lowered speed in top gear or at a higher speed in second gear. It is, of course, assumed that the tractor engine is governed not to exceed a designed maximum rotational speed.

The dynamometers are produced in batches on general purpose machines and are hand fitted on a method of selective assembly. All major components are of 3 per cent nickel steel. The smaller unit is made entirely from bar stock but in the larger model the end cylinders incorporating the coupling eyes are forgings. Plungers are hard chrome coated and ground to size with a limit of ± 0.001 in. The large phosphor bronze bushing in the sliding cylinder is pressed into place and is grooved for lubrication and finished to size in position. All seals and cup washers are of the graphited type manufactured by Ronald Trist and Co. Ltd. A refinement, not shown on the sectional drawings, is the provision of hardened steel bushings with radiused bores pressed in the coupling eyes in order to accommodate limited angular movement between tractor and trailer without stressing or loading the sliding joint. A leather bellows attached by Jubilee hose clips is fitted over the joint to exclude water or foreign matter.

Automobile lubricating oil, of SAE 30 grade, is used as the hydraulic medium. It is inserted by means of the screw-type gun provided through a filling orifice normally

closed by a ball held into a union cone seating by a cap nut. The system is bled at the union to the gauge. Immediately before the gauge a Budenberg damping valve is provided. In operation this valve is opened progressively only to an extent at which the needle of the gauge gives a steady indication of the load. Approximately one-quarter turn open is usually sufficient; too wide an opening will result in needle fluctuation. If the valve is closed when the load is first applied the possibility of the gauge being damaged by snatch loadings is obviated. While normally the gauge is mounted on the tractor vehicle, the instrument is frequently carried in the hands of a test engineer walking alongside the vehicle undergoing test.

Each instrument is given a shop test on completion of assembly. It is set up with a slave gauge in parallel with a comparator instrument, of known performance and fitted with an accurately calibrated gauge, and progressively loaded. After checking for conformance with the comparator unit and ability to hold all pressures, it is passed for dispatch to the N.I.A.E. There the instruments are carefully tested and the gauges are calibrated. They are subsequently returned to the works for the assembly of equipment and boxing. The net weights of the instruments are 22 lb and 54 lb for 10,000 lb and 40,000 lb models respectively, while the complete, boxed outfits weigh 74 lb and 104 lb respectively.

AIRLESS SPRAY PAINTING

STANDARD spray-painting involves the atomization of paint into fine particles by compressed air and blowing these particles upon the surface to be painted. Airless spray-painting accomplishes atomization by utilizing: (1) the mechanical force of hydraulic pressure released through a restrictive nozzle, and (2) vapour pressure, the paint being heated to such high temperatures that a fraction of the solvent content approaches the boiling point. The solvents do not boil because the circuit is under a pressure of 300-600 lb/in². When released to atmosphere, the solvents burst into gas, so contributing to break-up of the paint particles; the combined hydraulic and expansion effect achieves very fine atomization.

All types of finishes can be sprayed, and the material can be very thin or heavy-bodied, as desired. The solvent should preferably be a blend of high and low-boiling types. Film thickness is determined by the viscosity of the material, and heavy films are easily obtained, largely because more solvent is dissipated during atomization than is possible with conventional methods.

In the specially designed equipment, as indicated by a flow diagram, paint is siphoned in by a pump and passes through the heater to the spray gun; the unused portion is recirculated. Circulation is necessary to maintain hot paint at the gun nozzle at all times. A pressure-relief valve is fitted on the return side of the circuit. The spray gun is merely a trigger and is lighter than the conventional spray gun. The volume and angle of spray are controlled, not by gun adjustment, but by nozzle selection, so that the vital factor of balancing atomization does not depend on the operator.

Airless spray-painting considerably reduces both paint loss and the associated health hazard caused by rebound of the spray. In spray booths, the use of a slight exhaust to dissipate solvent fumes replaces the huge exhaust system formerly necessary. Finishes are superior because mixtures of higher and heavier solid content can be sprayed, and a greater percentage of solvent is dissipated during atomization. Adhesion is notably improved because the spray is hot and is not cooled down by the expansion of compressed air. *M.I.R.A. Abstract 55/12/41.*

IGNITION LAG IN DIESEL COMBUSTION

IGNITION lag in diesel-engine combustion is affected by inlet-air pressure and temperature, fuel temperature, jacket-water temperature and engine speed. Chemical delay appears to be rate-determining. Fuel volatility is less important than fuel structure. Additives used in concentrations of 0.5-1.0 per cent affect ignition lag, probably by chemical action.

Ignition lag is arbitrarily defined as the period elapsing between the start of injection and the time at which the pressure decrease caused by the cooling effect of the injected fuel has just been recovered. The components of ignition lag are injection delay, fuel-vaporization lag, and chemical delay.

It is concluded that adiabatic saturation is almost certainly approached very closely in the core of the spray. As distance from the centre of the spray increases, the air-fuel mixture becomes leaner, with consequently higher air-vapour temperatures. Under these conditions, adiabatic saturation is approached less rapidly. At the extreme edge of the spray, a few single droplets will probably be found. Closeness and rate of approach to adiabatic saturation conditions vary with distance from the core of the spray in different ways for fuels differing in viscosity and volatility. A volatile fuel does not receive heat as much more rapidly than a non-volatile fuel as would be expected from the differences in their volatility. Under adiabatic saturation conditions, a non-volatile fuel has at least as good a chance as a volatile fuel of achieving the combination of temperature and vapour-air ratio required for self-ignition and rapid combustion. Physical delay, so far from being a negligible portion of the total ignition lag, may actually be as large as, or larger than, the chemical delay period. Injection delay may not be insignificant in an operating engine. Different fuels vary somewhat in the way they receive heat following spray break-up, but the major differences between fuels of varying cetane number lie in the way they release chemical energy during the very early reactions. For the same fuel, total, physical, and chemical delays are smaller in an operating engine than in a combustion bomb operated at the highest temperature estimated to exist in the engine. *M.I.R.A. Abstract 55/12/22.*

NEW PLANT AND TOOLS

Recent Developments in Production Equipment

THE current practice of calling for only a short running-in period for new automobile engines has undoubtedly been to a great extent influenced by the increasing use of honing operations for finishing running parts. This practice is, of course, now common in this country, in the U.S.A. and in Germany. In fact, in Germany, honing is steadily replacing finish grinding operations on many components for which high surface finish or extreme accuracy are important.

Recently, some very interesting machines have been developed by Nagel in Germany. In all the Nagel machines, the movements are infinitely variable and hone diameters are adjustable by micrometer screw. There are brake drum superfinishing machines, which can be supplied with automatic transfer for fully automatic operation and with one or more spindles according to the output required. Automatic transfer machines for honing cylinder blocks are also built to special order.

Some typical Nagel machines are shown in Figs. 1, 2 and 3. Fig. 1 shows a type VM single spindle machine, incorporating an automatic sizing device, with a fixture for honing gears. Fig. 2 shows a similar machine with a special fixture for honing connecting rods. Four rods float in the fixture and the honing tools centre the components. Honing of the four rods is completed in 24 seconds. An auto-sizing attachment is applied to the machine. It is based upon a ring gauge which slides into the bore when size is reached. A two-spindle machine is shown in Fig. 3. It is set up for honing motor-cycle cylinder blocks. Wickman Limited, Coventry, are the sole agents for Nagel machines in the United Kingdom.

Gravity drop hammer

An improved type of gravity drop hammer, designed to meet the demand for more blows per minute, closer forging tolerances, reduced maintenance costs and greater safety in operation, is illustrated in Fig. 4. This new type of hammer, the Ceco-drop, is manufactured by the Chambersburg Engineering Company of America in a range of sizes from 500 to 10,000 lb capacity. It is intended to supplement the board drop hammer, which has reached a stage of efficiency which leaves little scope for further development.

The Ceco-drop incorporates a steam or air-lift piston rod unit which replaces the boards of a board hammer. This gives a higher lifting speed and, therefore, more blows per minute. The piston rod is considered to be expendable, and it was originally estimated that a working life of 200 operating hours would be more economical than the boards of a board hammer. However, a careful check on the first twenty Ceco-drops put into service proved that an average life of 800 hours was obtained. Many rods have been in service for as long as 1,200 hours.

The illustration shows the largest Ceco-drop yet built. It is rated at 8,000 lb capacity, and is to be installed in the Sheffield plant of the English Steel Corporation Ltd. This gravity fall hammer weighs 121 tons, has a stroke of 54 in and will strike a maximum blow of 45,000 ft-lb energy. The main parts of the hammer, the anvil, frames and yoke, are steel castings. They were cast and machined by the English Steel Corporation Ltd. Erection of the hammer was carried out by the Davy and United Engineering Co. Ltd., Sheffield, who build Chambersburg equipment, under

licence, in this country. Alfred Herbert Ltd., Coventry, are the sole agents for the Chambersburg Engineering Company.

Centreless grinder

The most recent addition to the range of grinding machines designed and manufactured by Cincinnati Milling Machines Ltd., Birmingham, 24, is the No. 2 centreless grinder illustrated in Fig. 5. This machine incorporates certain features, notably a fixed grinding wheel spindle mounting and Filmatic spindle bearings, which have proved successful on many other Cincinnati grinders. Additionally, several new and important features are incorporated in the design.

One of the interesting developments is the arrangement of the slides. A pre-loaded precision ball bearing lower slide is mounted on bed ways at right angles to the grinding wheel spindle. It supports an upper slide unit which carries the regulating wheel. This form of construction simplifies setting-up and sizing adjustments, because the work rest is mounted on the lower slide, where it can be readily positioned in the correct relation to the two wheels. The way bearing surfaces are well guarded to exclude grit and cutting fluid. To compensate for slight errors in truing and/or set up, the complete regulating wheel pile may be swivelled slightly on the bed.

The regulating wheel is driven by a built-in 1½ h.p. motor through a back gear combination and a wide vee belt. The belt rides on four variable pitch, cone-shaped sheaves, and regulating wheel speeds are infinitely variable in two ranges, 13 to 78 and 68 to 392 r.p.m. Speed changes are selected by a handwheel and indicated on a large tachometer dial.

Power-operated profile truing for the grinding wheel is standard equipment. It is actuated by a hydraulic motor at one end of the unit. Two external truing controls are located on the operating side of the bed. One is for engaging rapid and truing rates, and for changing the direction of traverse; the other is for changing the rate of traverse. The truing unit is designed for formed cylindrical cams. To exclude dust, the truing slide is efficiently guarded. On the standard machine, regulating wheel truing is effected manually through a screw type unit, but power-operated truing is available as an extra. Both units are arranged for diamond truing.

On this new machine it is a simple matter to reset a trued regulating wheel. After truing, it is merely necessary to unclamp the upper slide of the regulating wheel housing and move it in an amount equal to that removed from the wheel radius. This brings the wheel into the before truing position. The regulating wheel guides do not have to be re-adjusted after the regulating wheel has been trued.

All the controls have been centralized at the operator's normal working position. The conveniently located in-feed handwheel for size adjustment and positioning the slides is on the upper slide. It incorporates all the refinements of a centre type grinder cross feed mechanism, with two dials graduated respectively to 0.0001 in and 0.001 in. For in-feed grinding the operator does not use the handwheel, but rather the in-feed lever connected directly to the in-feed screw. A 90 deg arc of this lever, downward and return, completes the manual in-feed cycle.



Fig. 1. Nagel single spindle honing machine

Fig. 2. Honing connecting rods on a Nagel machine

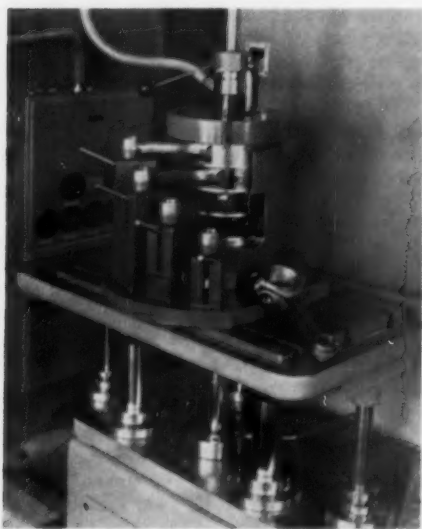


Fig. 3. Two-spindle Nagel honing machine

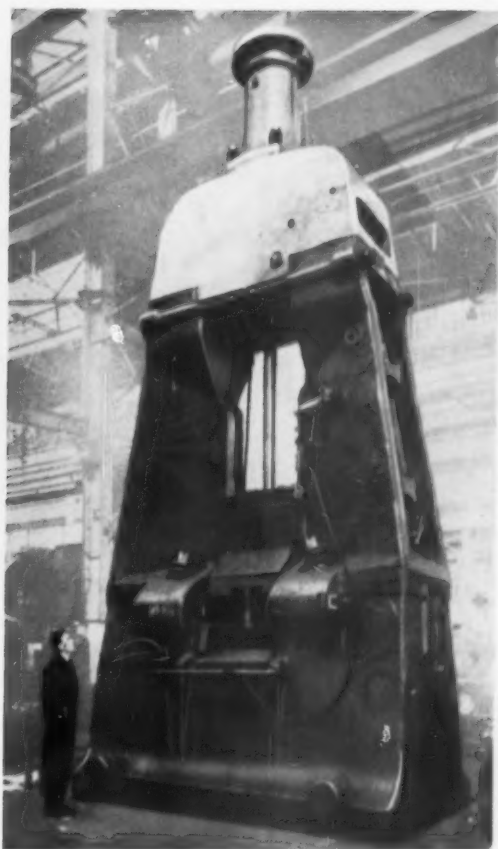
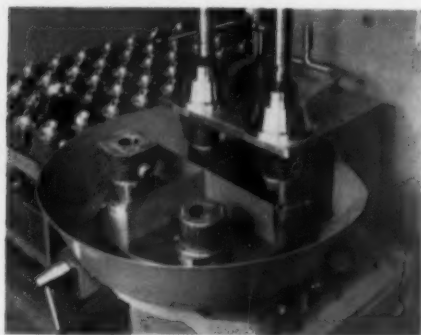
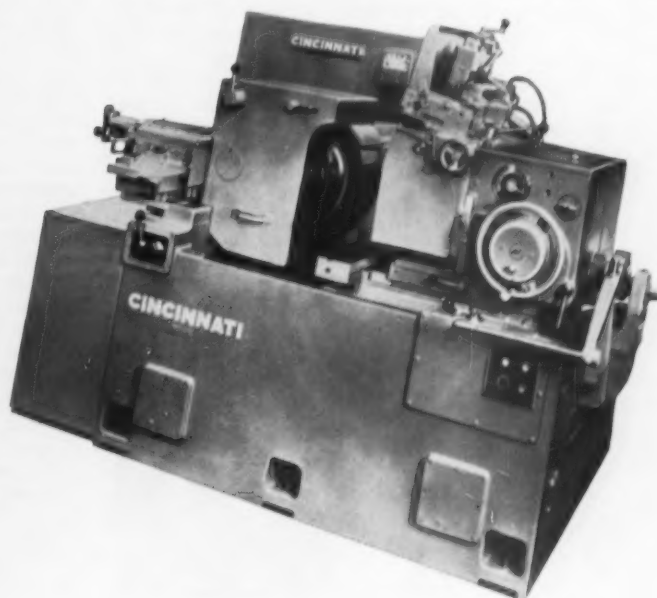


Fig. 4. Chambersburg 8,000 lb capacity gravity drop hammer

Fig. 5. The new No. 2 Cincinnati centreless grinding machine



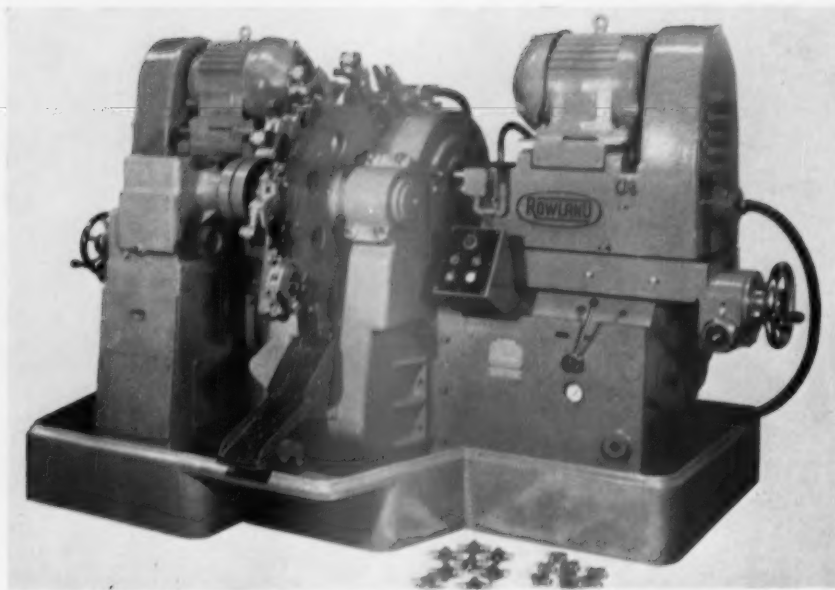


Fig. 6. Rowland duplex surface grinder for cruciform spiders of universal joints. High output rates and high precision are obtained from this machine

Duplex grinder

F. E. Rowland & Co. Ltd., Climax Works, Reddish, Stockport, are continuing their development work in connection with their duplex surface grinders, and a recent example is a machine for grinding simultaneously the opposite ends of the cruciform spider of the universal joint generally used for motor vehicle propeller shafts. The machine illustrated in Fig. 6 is of particular interest as not only must the overall length of the two trunnions be ground to size, but they must be square with the opposite trunnions, and also the respective bearing portions must be concentric to each other to very close limits, whilst at the same time a high rate of production is necessary on a continuous basis. The machine is built in two sizes, namely 30 in and 20 in.

These machines are built to J.I.C. standards and are based on a substantial fabricated steel bed, on which are mounted massive cast iron slideways carrying the two grinding wheelheads, each driven by its own independent motor mounted on the top. Both wheelheads are adjustable in both the horizontal and vertical planes in order that the grinding wheels may be set at an angle relative to each other and to the work carrier, in order to provide the most efficient grinding conditions to obtain the particular accuracy called for in the case of each specified component. The grinding wheels are of the inserted nut type of abrasive disc, and are

mounted on massive steel disc backplates by a series of screws, and these are carried on a substantial cast iron backplate or flange on the ends of the spindles. Balancing weights on the disc plates enable the complete plate and grinding wheel to be accurately balanced before mounting. The high tensile steel grinding wheel spindles are of exceptionally large diameter and run on pre-loaded angular contact bearings at the wheel end, giving accurate location with a generous thrust capacity. The pulley end is carried on a roller bearing in such a manner that temperature rise does not affect the accuracy of the component being ground.

The wheelheads are operated by hydraulic power, which maintains them against special adjustable stops which are provided with coarse and fine feed—the former for rough setting and the latter for sizing. Hydraulic power is provided by a separate pump and tank unit including motor drive, relief valve and pressure gauge, and is in accordance with J.I.C. provisions.

Drive to the rotating work carrier is by a motor through a vee belt and a worm reduction gear, a friction clutch being incorporated to ensure no damage results should oversize components be fed into the carrier. Provision is made for varying the speed of rotation. On the periphery of the carrier are mounted a series of special fixtures very accurately located, carrying the two trunnions to be ground in vee blocks; the component is firmly held in position by a curved cam clamping bar as it passes between the grinding wheels. The work is loaded by hand and when released by the clamp, is delivered to the chute at the base.

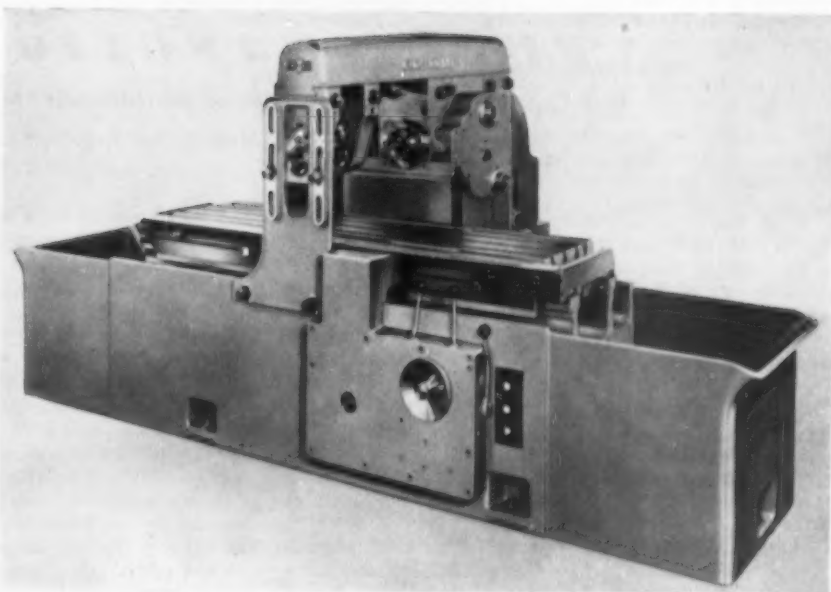
Compensation for grinding wheel wear in order to maintain size is by suitable adjustment of the special dead stops at each end, but a hydro electric compensation device is available whereby the necessary increments of feed or cut are obtained by operation of a convenient push button.

A substantial grinding wheel dressing attachment is provided, as shown in Fig. 7. It can accommodate either diamond tools or cutters, according to requirements, and the sliding parts are protected from wear by abrasive dust by adequate covers. The dresser is hand-operated in the case of the 20 in machine and by hydraulic power in the case of the 30 in machine. A rapid traverse motion to the grinding wheelheads for wheel changing and also wheel dressing is provided. It is operated by the long control lever to be

Fig. 7. Grinding wheel dressing attachment for the machine shown in Fig. 6



Fig. 8. Cincinnati HyPowermatic milling machine. HyPowermatic machines have intermittent, dog controlled automatic table cycles and infinitely variable feed rates



seen in the illustration below the right-hand grinding wheel-head. The machine is provided with an adequate coolant system by supplies through both hollow spindles and a central nozzle from a motor-driven suds pump—whilst a combined magnetic and paper filter is available.

The driving motors are of 15 h.p. for the 20 in machine and of 25 h.p. for the 30 in size. In both cases the whole of the starting gear for all motors and solenoid valves is contained in a floor mounted panel which is operated by a series of push buttons contained in a "desk" type panel at the operating position. This includes warning and indicating lights. As it is impossible during working to see how far the grinding wheel has worn, a device is incorporated warning the operator by means of a red light that the wheel is at the end of its useful life and that replacement is necessary. Production is possible at rates of up to 1,600 pairs of ends per hour, depending on size, and the faces are held square to the axis to within 0.0005 in and concentricity with the locating axis within 0.0005 in, whilst uniformity for size is maintained to within 0.0015 in.

Bed-type milling machines

A new line of heavy-duty bed-type milling machines, see Fig. 8, with a new name—HyPowermatic—has recently been developed by Cincinnati Milling Machines Ltd., Birmingham. The machines are equipped with automatic two-way feed cycles and infinitely variable feed rates, and are designed for continuous operation on medium to larger size parts. Much heavier and more powerful than the superseded models, the new machines offer increased cutting capacity (up to 50 h.p.) and higher spindle speeds (up to 2,000 r.p.m.) for taking conventional milling or climb milling cuts with either high speed steel or sintered carbide milling cutters. Standard machines, designated the 300, 400 and 500 series, are built in plain and duplex styles in forty-two sizes of each, from 36 in table travel, 7½ h.p., to 168 in table travel, up to 50 h.p.

HyPowermatics are provided with intermittent, dog controlled automatic table cycles. Feed rates may be infinitely varied throughout their complete range of ½ in to 100 in or 150 in per minute (depending upon machine size) by means of an easily operated feed rate selector dial. The table is driven by a new "Hydramech" type of unit, which

is enclosed within the bed where it is protected from dust and grit. It consists primarily of a hydraulic motor, with an infinitely variable arrangement, driving a worm and dual worm which, in turn, drives twin vertical pinions engaging the table rack. Anti-friction bearings throughout and automatic pressure lubrication system assures long life and trouble-free service.

Operating controls are conveniently and compactly grouped on the front right-hand side of the bed at the operator's normal working position. Quick positive starting and stopping of spindle rotation is obtained through two hydraulically operated multiple disc clutches. Both are actuated from the spindle start-stop lever at the operator's station. Also included in the control group are the four-position directional control lever to provide engagement of the table feed and rapid traverse (right and left), the table stop, and automatic spindle stop levers. Sixteen spindle speeds can be obtained through change gears and a back gear combination. Nine ranges of spindle speeds are available. The higher group ranges from 50 to 2,000 r.p.m. for 300 Series spindle carriers, 30 to 1,200 r.p.m. for 400 Series spindle carriers and 20 to 800 r.p.m. for 500 Series spindle carriers.

Lubrication of the principal units of the machine is completely automatic. Table ways and the drive mechanism are lubricated by a power driven pump from a self-contained reservoir within the bed. In addition to the conventional sight gauge, the reservoir is equipped with a float switch to stop the drive motor automatically if the oil should be exhausted below the low limit. Spindle bearings, gears and other parts within the carrier are automatically pressure lubricated, while the arbor bearing collars have gravity lubrication from a reservoir in the arbor support.

Electrical controls and push buttons are built-in, protected against damage, dust and moisture. To eliminate electrical hazards, the hinged door for the control compartment is equipped with a built-in mechanically interlocked disconnect switch. There are other electrical safety features, too. A small contact button built into the change gear compartment automatically stops the main drive motor when the hinged cover is opened. This feature protects the operator from rotating gears should he forget to press the main stop button when changing spindle speed pick-off gears.

FUEL INJECTION

Metering Control with Timed Injection as an Alternative to Carburation

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SINCE the automobile piston engine is a type of heat engine, deriving its power by conversion of energy released in the working fluid, its performance depends primarily upon the rate of fluid throughput, the energy content of the fluid, and the efficiency of the conversion. The energy conversion is, of course, effected by combustion and expansion within the cylinders, and the efficiency is determined largely by factors of engine design. Direct control of the engine performance can be exercised by selection of the quantity and quality of the fluid flow. In most instances, a simple master control in the form of a throttle valve is employed to regulate the air mass flow, and the fuel mass flow is made dependent upon the air flow so that mixtures of the required strength are delivered to the cylinders.

Although carburetors have been used for many years to provide the necessary fuel-air flow relationship, the introduction of the fuel into the cylinders, instead of into the manifold, offers many advantages. This article is confined mainly to the linking of the fuel and air flow controls for timed fuel injection systems for spark ignition engines.

Carburation

Before embarking on a study of injection problems, it is necessary to have a clear knowledge of the fundamental principles of carburation, so that the direction in which improvements may be made can be discerned. The process of feeding the correct mixture to a piston engine running under varying conditions involves primarily the measurement of the air flow so that the appropriate quantities of fuel can be metered into it. In principle, the simplest instrument available for this purpose is a carburettor, comprising a venturi tube for air flow measurement and a fuel metering jet with a delivery orifice in the venturi throat, the fuel flow being controlled by the constant supply pressure to the jet and the variable depression in the venturi. The mixtures required by a carburetted engine include a very rich mixture for cold starting, a fairly rich mixture for slow running, a weak mixture for cruising, and a rich mixture for full throttle power, Fig. 1.

Because of the natural laws that operate in a simple carburettor of the type described in the previous paragraph, the mixture is progressively enriched as the air flow is increased, and no fuel at all is delivered below a certain critical minimum air flow. This supply metering characteristic must therefore be tailored to suit the demand characteristic of the engine, hence the need for such additional devices as starting chokes, compensating tubes, power jets, economizing valves, and so on. Since, in practice, the constant fuel pressure at the metering jet is controlled by means of a float chamber, the carburettor is sensitive to gravitational effects. Another defect of the system is that a momentary weakening is caused by sudden opening of the throttle; this is because of the dependence of fuel flow upon the air flow. An accelerator pump may be necessary to obviate a flat spot during acceleration. The evaporation of the fuel introduced into the venturi causes a drop in temperature. This drop may be as much as 25 deg C; under severe winter conditions, ice may form in the carburettor and tend to block the path of the incoming air, and even to lock the throttle.

One means of overcoming many of these shortcomings of the conventional carburettor is to replace the constant-

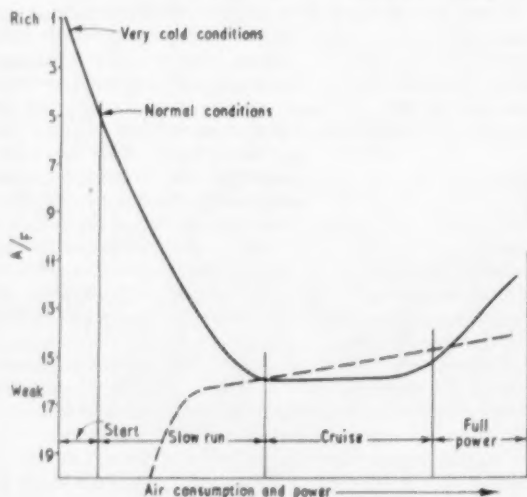
head float chamber by a diaphragm system controlled by the pressure of the flowing air. To control the rate of fuel flow, sources of high and low air pressure are required. This arrangement not only eliminates the gravitational effects, but it also enables the fuel to be introduced into the manifold on the engine side of the throttle so that carburettor icing does not arise.

Fig. 2 illustrates the principle of the pressure carburettor, as applied in the Bendix-Stromberg instrument for aircraft engines. Two diaphragms are used; they are connected by a rod, which forms the needle that operates in the fuel supply jet. The difference between the impact and venturi pressures acting on each side of the air diaphragm is balanced by the difference between the pressures on the two sides of the fuel diaphragm, that is, the pressure on the fuel supply side of the system and that on the delivery side of the metering jet. In this way, the metered fuel flow is controlled by the air flow.

With these complications and modifications to design, the performance of the erstwhile simple carburettor can be made to match the engine requirements. The metered supply of fuel, however, must traverse an appreciable length of manifold before it reaches the cylinders. Moreover, the mixture stream must also negotiate a number of manifold bifurcations and bends, each of which may disturb the distribution of fuel over the cross sections of the passages.

It is generally recognized that the air and vaporized fuel mix reasonably uniformly. However, the droplets of liquid fuel are either airborne or are blown along the manifold walls; thus, their distribution depends largely upon inertia effects and the geometry of the manifold. As a result, the distribution between individual cylinders may differ by as much as ± 2 units of air:fuel ratio, so that the power output from some of the cylinders is low, while in others fuel is wasted. This is particularly serious when knocking conditions are reached, since premature knocking in only one

Fig. 1. Mixture requirements of carburetted petrol engine
--- Carburettor characteristic before compensation



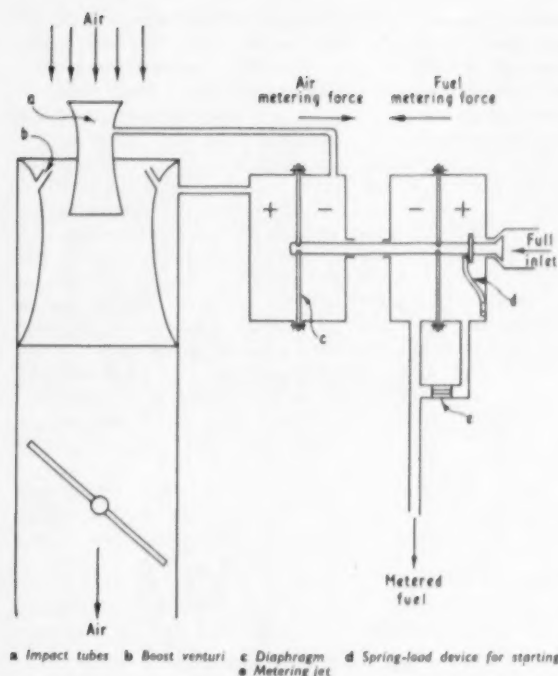


Fig. 2. Principle of the Bendix-Stromberg pressure carburettor for aero engines

cylinder will limit the performance of the whole engine.

To improve distribution, induction manifolds have been developed with smooth internal surfaces to facilitate flow, and with sharp right-angle bends to induce the surface travelling fuel to return into the air stream. Hot spots and thermostatically-controlled manifold heaters are also employed to reduce the quantity of unvaporized liquid fuel. In addition, suppliers of the fuel ensure that the fuel volatility is maintained at a high level, consistent with the limitations imposed by the possible onset of vapour-lock troubles.

It is clear that optimum performance involves smooth running with equal power from each cylinder; therefore uniform distribution of the mixture and identical sets of other controlling variables, such as valve timing and ignition timing, are essential. Timed fuel injection, either through the inlet ports or into the cylinders, offers a means of obtaining complete control of fuel distribution and from this viewpoint is most attractive. With this system, the mixture supply requirements are not influenced by fuel evaporation in the manifold, so the enrichment for starting and slow-running need not be so heavy. With timed injection, the greater uniformity of distribution permits the use of weaker mixtures. The engine mixture requirement curve is therefore of the same general form as that in Fig. 1, but lies at a slightly higher level of air:fuel ratio.

As a first step, the pressure carburettor shown in Fig. 2 could be used as a master control for a timed, instead of a continuous, fuel supply. In fact, the Bendix unit has been applied successfully to a direct cylinder-injection system on the Wright R-3350 series 18 BD supercharged aero engine. A differential diaphragm unit is fitted to the injection pump, so that the pressure of the metered fuel arriving at the pump controls the effective plunger stroke in such a way that the delivery to and output from the pump are exactly matched.¹

However, it is desirable to control simply and reliably the output of the individual pump elements of a timed injection system by some method that avoids the inherent

drawbacks of the carburettor venturi. With this aim in view, alternative methods, in which the engine itself is used as the air meter, have been developed.

Speed-density metering

The rate of mass flow of air into the cylinders, with which the fuel flow is to be matched, depends upon the volume swept by the pistons per unit time, the density of the air at the inlet ports and the ability of the air to occupy the swept volume; that is;

$$W_a = n V_s \frac{N}{2} \rho_i \eta_v$$

where W_a = air mass flow rate in lb/min

N = r.p.m.

n = number of cylinders

V_s = swept volume of one cylinder

ρ_i = density of air at inlet ports

η_v = volumetric efficiency, defined as the ratio of the weight of air actually taken into the cylinder to the weight of air which would exactly fill the swept volume at the inlet density

This expression can be reduced to $W_a \propto N \rho_i \eta_v$.

As a constant valve overlap has to be used, the volumetric efficiency rises as ρ_i increases, and falls as N and the exhaust back pressure increase. Since these changes are slight, a constant average value of volumetric efficiency can be assumed, for example, approximately 85 per cent, and the expression simplified to $W_a \propto N \rho_i$. This is termed the *speed-density* expression and is based on the assumption that the mass of air consumed by the engine is directly proportional to the engine r.p.m. and to the air density in the inlet manifold.

In carburation, the matching of the supply mixture-characteristic with that of the engine is carried out in a two-stage process comprising compensation to give a constant economic mixture, and enrichment as required. In a similar way, the primary duty of the control unit of a fuel-injection pump is to maintain a constant mixture and to enrich it only when the demand arises. The fuel flow must therefore follow a law similar to that of the airflow, and the speed-density expression can be used as a basis for the pump control unit design, that is $W_f \propto N \rho_i$. By using the characteristic equation $PV = RmT$, this relationship can also be expressed as $W_f \propto \frac{NP_i}{T_i}$

where P_i and T_i are the absolute pressure and temperature respectively at the inlet port. However, in practice it is found that the mechanism required to give a reciprocal temperature effect is excessively complicated, and a negative effect is generally accepted to avoid undue complexity. The fuel control device therefore follows the law $W_f \propto N (P_i - kT_i)$, and the initial setting is chosen to minimize the errors incurred. The direction of these errors is such as to give over-enrichment both at low air temperatures, which in practice is useful for starting, and at high temperatures, which helps to avoid overheating.

Since timed-injection pumps are engine-driven, and injections are synchronized with the induction strokes of the pistons, the speed factor is automatically catered for and the fuel flow rate increases and decreases with r.p.m.. A change of speed at constant throttle position varies the pressure in the manifold, but this again is allowed for in the case of a speed-density type of control unit. For example, a reduction in speed at constant throttle increases the air pressure in the manifold and hence the air consumed per induction stroke. The control system, being engine-driven, would reduce the fuel flow per unit time and, since it is also density sensitive, would increase the fuel flow per injection and would thus maintain a constant mixture strength. An increase in air density due to a reduction in air temperature would have a similar effect on the fuel

flow per injection into the cylinders of the engine.

S.U. fuel injection pump

In the S.U. system², each injector is supplied by an individual pump element of variable stroke, driven by a swash plate and Z-shaft mechanism, Fig. 3. The Z-shaft is mounted on the main shaft, and both rotate together. A servo mechanism is employed to move the Z-shaft axially along the main shaft, to change the attitude of the swash plate and thus to alter the fuel delivery. The servo piston also forms the bearing that carries one end of the Z-shaft. Its axial position of equilibrium is determined by the engine-oil pressure acting on the piston face nearest to the plungers, and opposing a reduced oil pressure together with a spring load acting on the other face of the piston. The main shaft is hollow, and the flow of engine oil through to the control chamber is determined by the position of the servo valve relative to its seat on the end of the main shaft. A restriction is provided in the oil outlet from the chamber that houses the control spring, so that the oil pressure in the chamber does not fall below a level sufficient for the maintenance of effective control.

A nitrogen-filled capsule stack, sensitive externally to manifold pressure, is maintained at manifold temperature by the continuous passage over it of air from the manifold. The capsule stack is directly linked with the servo valve in the control chamber, and the control spring connects the

obtained by rotating the elements to bring the spiral cut-off groove in line with the relief port³. Again, since the pump is engine-driven, the fuel output is automatically regulated to suit the speed. A vacuum connection is made from the manifold to one side of a control diaphragm fitted on the end of the pump rack. Manifold depression acting on one side of the diaphragm is opposed by a compression spring on the same side, Fig. 4. An increased manifold pressure, that is, a reduced manifold depression, permits the spring to move the diaphragm and the rack in the direction that gives greater fuel output. Since atmospheric pressure acts on the control diaphragm, compensation for changes in atmospheric pressure has to be effected. To this end, a capsule stack is connected through a linkage to the rack, and variations in atmospheric pressure on the outside of the capsule apply an additional force to the rack. For example, an increase in atmospheric pressure which would not affect the control diaphragm directly because of the rise in pressure on both sides, compresses the capsule and moves the rack so that more fuel is delivered.

Since the manifold temperature varies in a similar way to atmospheric temperature, and since the capsule is filled with air, changes in atmospheric temperature also exert a controlling force through the linkage. Thus, correction is made to cater for the negative temperature term in the expression for fuel flow. A hand-controlled, mechanical over-ride is provided to move the rack into the maximum

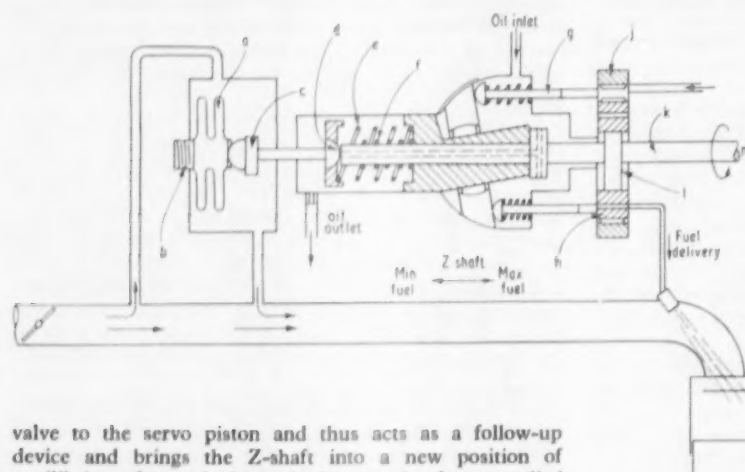


Fig. 3. Left: principle of the S.U. fuel injection pump control system. In automobile applications, the pump is mounted vertically with the plungers uppermost, the capsule chamber is mounted on the side of the unit, and there is a lever connection to a sleeve type of servo assembly

a Capsule containing nitrogen b Adjustment c Ball race d Servo valve e Control spring f Enrichment spring g Plunger h Geared distributor valve i Internally toothed fixed gear ring k Mainshaft l Eccentric m Drive from engine

valve to the servo piston and thus acts as a follow-up device and brings the Z-shaft into a new position of equilibrium after each change in the metering forces applied by the capsule member. An increase in manifold pressure, for example, would compress the capsule, open the valve, increase the oil pressure in the control chamber, and move the Z-shaft in the direction of higher output. This movement would tend to close the valve again, by relieving the compression of the control spring, and so re-establish equilibrium at the higher fuel output. A reverse process would take place with reduced manifold temperature.

In the absence of any oil pressure, that is, for starting, a rich mixture is provided by the action of the control spring. At high manifold pressures, enrichment is obtained by the action of two control springs: the inner spring goes out of action at high manifold pressure, so that the spring rate is effectively reduced, and the Z-shaft has to move a greater axial distance to give an equivalent load effect on the servo valve. The spring rate of the capsule stack is designed to give a rapid response for acceleration.

Bosch fuel injection pump

This follows normal diesel practice, in that the stroke of the individual pump elements is constant, the timing of the start of injection is fixed, and a variable cut-off is

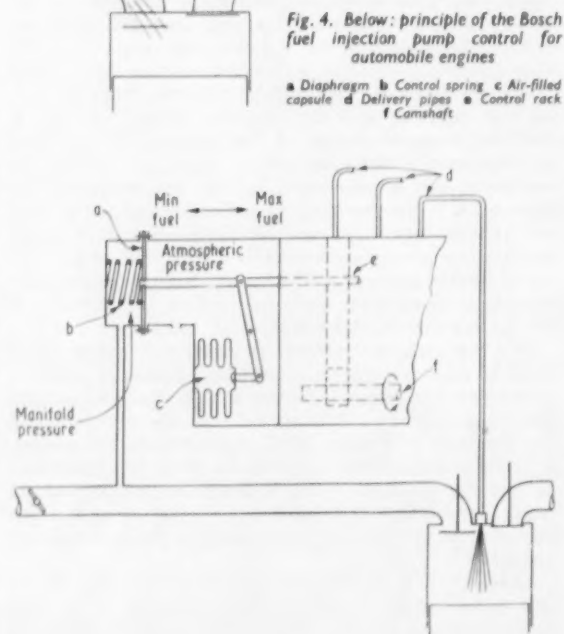


Fig. 4. Below: principle of the Bosch fuel injection pump control for automobile engines

a Diaphragm b Control spring c Air-filled capsule d Delivery pipes e Control rack f Camshaft

flow position for starting, and a number of compensating holes are automatically selected throughout the range of throttle movement to influence the depression in the vacuum pipe and give the required degree of enrichment.

Speed-density-back-pressure metering

When the exhaust back pressure varies appreciably and independently from the inlet pressure, as with supercharged engines operating at varying altitudes, the volumetric efficiency term in the expression for air consumption can no longer be ignored. The reduced back pressure at altitude improves the scavenging from the cylinders, and leads to a greater air throughput, but, of course, does not alter the theoretical air capacity of the cylinders. In these circumstances, the fuel pump control device must be made sensitive also to back pressure; it must provide for increased fuel flow as the back pressure falls.

The expression for fuel requirement can be developed by subtracting the mass of the gases contained in the cylinders at bottom dead centre at the end of the induction stroke from the mass of the exhaust residuals contained at top dead centre at the end of the exhaust stroke. With injection into the cylinder, the incoming charge consists of air only, the fuel being added later, whereas in the case of injection into the port, both fuel and air are inhaled. Since a constant fuel:air ratio by weight is required primarily, the theory developed in the next paragraph applies to both cylinder and port injection.

At bottom dead centre after induction, the induced charge is assumed to be at the pressure level P_1 of the inlet manifold and to have been warmed, by the residuals, from the inlet manifold temperature level, T_1 , to some intermediate level, T' . The residuals are assumed to be at the exhaust manifold pressure and temperature, P_E and T_E respectively. A problem arises with regard to the term T' , since it is transient and is difficult to calculate or measure. By ignoring the heat given to the charge from the cylinder walls, and assuming that the heat lost by the residuals equals the heat gained by the charge, and also that the specific heats are equal, the term T' can be eliminated. The expression for the charge mass flow rate then becomes:

$$W_c = \frac{nN(V_c + V_e)(P_1 - P_E/c)}{2RT_1} \text{ lb/min}$$

where W_c = charge mass flow rate

V_c = clearance volume of one cylinder

R = characteristic constant = 95 ft-lb/lb-deg C for charge

c = engine compression ratio

This is known as the *charge-weight law*, which can be simplified to the *speed-density-back-pressure* expression, and applied to the fuel mass flow rate, as shown:

$$W_f = \frac{P_1 N (P_1 - P_E/c)}{T_1}$$

This can be further modified to:

$$W_f = \frac{P_1 N (P_1 - P_E/c - kT_1)}{T_1}$$

to give the more simple mechanical control for temperature.

The back pressure term, being negative, can easily be accommodated by arranging that the back pressure acts upon the capsule stack in the same direction as the inlet temperature. Since this term is divided by the compression ratio, the area of capsule surface on which the back pressure acts is restricted in the appropriate proportion, Fig. 5. The exhaust back pressure is usually a constant small value, for example, 1 lb/in² above ambient, so in aircraft practice it is convenient to use ambient pressure instead of tapping from the exhaust manifold.

Hobson fuel injection pump

This pump incorporates constant-stroke pump elements, and on each plunger two spiral grooves are arranged so that the timing of the centre of the injection period is constant

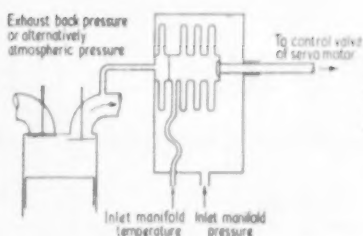


Fig. 5. Capsule stack modified for back-pressure to act on a reduced area of capsule

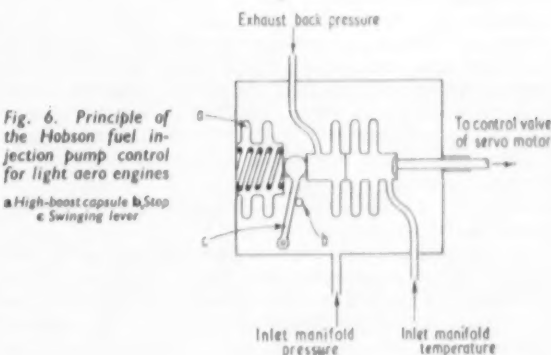


Fig. 6. Principle of the Hobson fuel injection pump control for light aero engines

a High-boost capsule b Stop c Swinging lever

and both the beginning and end of injection can be varied with movement of the control shaft⁴. A capsule stack and oil-operated servo system are used for regulation, on the principles already described. In addition, a negative-action, reduced-area capsule, subjected to back pressure, is incorporated. With supercharged engines, heavy enrichment is required, under full power conditions, to provide internal cooling and prevent knock. This is obtained by bringing into operation a second capsule stack, in series with the first, so that the manifold boost pressure acts upon a greatly increased capsule area and has a correspondingly increased effect upon fuel flow, Fig. 6.

Conclusions

Although the relative merits of fuel injection and carburation have not yet been outlined in detail in this discussion, a number of them have become apparent. With fuel injection in supercharged engines, the lack of fuel cooling effect in the supercharger is a serious drawback, and may be sufficient to offset the increase in power that would otherwise be obtained. However, in general, the advantages of uniform distribution, easier starting, and the elimination of backfire hazards, make fuel injection attractive. Continuous injection on a speed-density-back-pressure basis has almost completely replaced carburation in aviation, and speed-density timed injection is increasing in popularity for racing car engines. High costs and the complication of fuel injection equipment have retarded a general application in the automotive field, but the demands for higher performance are insistent, and the many problems of simplification and adaptation for large-scale production may yet be solved.

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PERFORMANCE PREDICTION

A Simple Graphical Method for the Determination of Vehicle Acceleration

IN an article published in the September 1954 issue of the *Automobile Engineer*, attention was drawn to simple, accurate and time-saving graphical methods of vehicle acceleration determination, due to Desdoutis (France)—Lomonosoff (Russia) and Lipetz (Russia)—Strahl (Germany). These are methods for rapid determination of the speed-time and speed-distance curves, which in turn provide the data for plotting the vehicle performance in terms of time-distance curves. Such curves are of considerable interest to operators. A direct determination of time-distance data can be carried out by a novel and straightforward procedure suggested by Prof. W. Müller.¹ This method has a number of advantages, among which are simplicity, clarity and time saving.

As an illustration of the method, an analysis of the performance of a single-deck bus will be made as follows. The weight of the vehicle is 10.75 ton fully laden, and it has a 130 b.h.p. engine. Its frontal area is 72 ft². The transmission is of a hydro-mechanical type, comprising a torque converter and two gears,² the gear ratios being 1:1 and 3:1. The high ratio is used for operation over hilly routes, and direct gear is employed in normal city service. A 15 per cent loss in efficiency through the transmission is assumed; the resultant tractive effort curves, together with the resistance curve for the vehicle running on a smooth level road, are plotted in Fig. 1.

In the following, the tractive effort as developed in the "normal" gear only will be considered. Since force=mass times acceleration, $p=m \times a$, and $a=dv/dt$, the force is determined from $p=m (dv/dt)$. For the step by step integration, the difference values Δt seconds and Δv ft/sec are substituted for dt and dv respectively, and a suitable constant value for the time increment Δt is selected. This gives the value of Δv , which $=v_2 - v_1$, where v_1 is the known velocity at the beginning of the time increment Δt , and v_2 is the velocity at the end of that increment. The effect of the inertia of the rotating masses such as the wheels, shafts and transmission gears must be added to the mass m of the vehicles. This effect can be accounted for by adding about 5 per cent to the value of m .

The units employed are as follows: $m=W/g$, where W is the vehicle weight in tons and g the gravitational acceleration in ft/sec², the velocity V is expressed in m.p.h. and the data of Fig. 1 is used in terms of excess tractive effort p lb/ton, that is, tractive effort less tractive resistance available on the level for acceleration and negotiating gradients, as shown by the shaded area in Fig. 1. It follows that

$$p = \frac{2240 \times 1.05 \times 1.467}{32.17} \times \frac{\Delta V}{\Delta t}$$

$$= 107 \frac{\Delta V}{\Delta t} \text{ lb/ton}$$

A 45 deg set-square can be used to effect the time-distance determinations easily and quickly. To do this, the mean

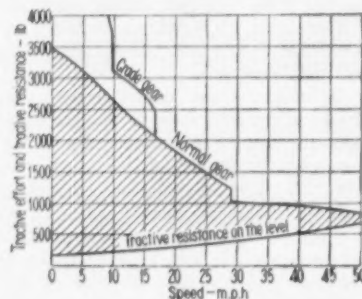


Fig. 1. Tractive effort and resistance curves for a vehicle running on a smooth level road

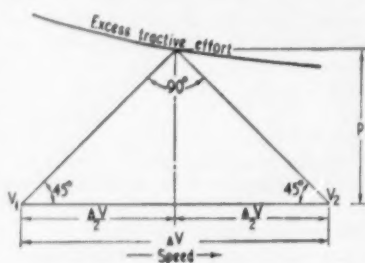
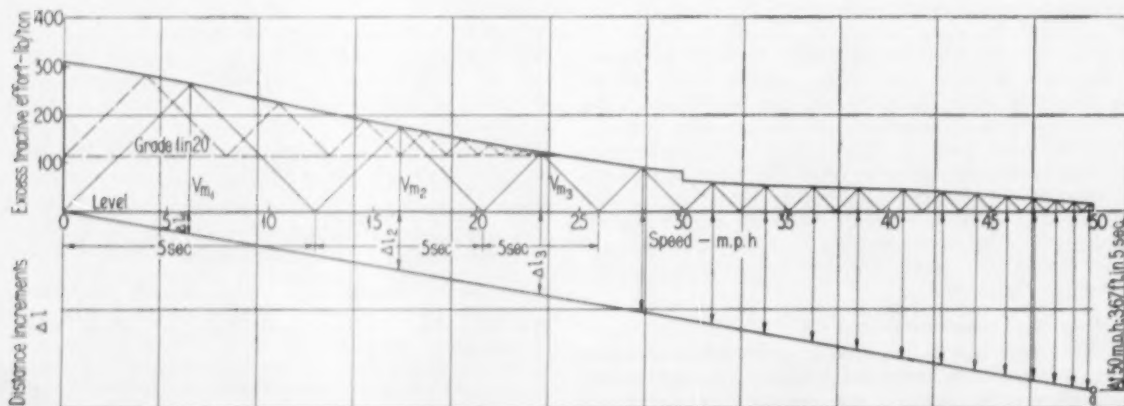


Fig. 2. Diagram showing how the mean value of p for each time increment Δt can be taken as that obtaining mid-way between the beginning and the end of each increment of vehicle speed, ΔV

Fig. 3. At the arbitrary speed of 50 m.p.h., the distance covered during the time Δt is 367 ft and the time taken to cover that distance is 5 sec. This is plotted to an arbitrary scale below the horizontal axis to give the point a , which is connected by a straight line to the point of origin



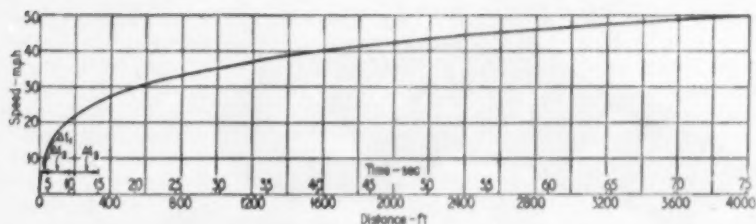


Fig. 4. The values of Δt and ΔV of Fig. 3 are plotted along the base of this curve so that the distance covered in any time can be seen at a glance. In addition, the vehicle speed is plotted as a function of both distance and time

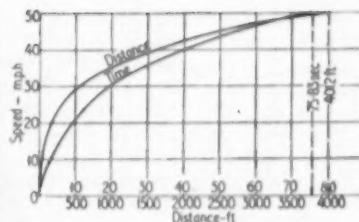


Fig. 5. Curves determined by the conventional analytical method to check the results obtained in Fig. 4

value of p for each time increment Δt is taken as that obtaining mid-way between the beginning and the end of each increment of vehicle speed ΔV , Fig. 2. Thus

$$p : \Delta V/2 = \tan 45 \text{ deg}$$

$$= 1$$

$$\text{and } p : \Delta V = \frac{1}{2}$$

Therefore

$$\frac{p}{\Delta V} = \frac{1}{2} \\ = \frac{107}{\Delta t}$$

If the excess tractive effort scale selected is: $p = 100 \text{ lb/ton} = 1 \text{ in.}$, or $1 \text{ lb/ton} = 0.01 \text{ in.}$, the velocity scale of $V = X$ is determined from:—

$$\tan 45 \text{ deg} = 1$$

$$= \frac{p \times 0.01}{X \times \Delta V/2}$$

$$\text{or } X = 0.02 p / \Delta V$$

so that for an arbitrary value of $\Delta t = 5 \text{ sec}$

$$\frac{p}{\Delta V} = \frac{107}{5} \\ = 21.4$$

and $V = 1 \text{ m.p.h.} = X = 0.02 \times 21.4 = 0.428 \text{ in./m.p.h.}$

The curve of excess tractive effort against speed is plotted to this scale in Fig. 3. At an arbitrary speed, in this instance 50 m.p.h., the distance covered during the time Δt , that is, 367 ft in 5 sec, is plotted to an arbitrary scale below the horizontal axis through the origin, and the point a is connected by a straight line to the point of origin. The distances Δl between this line and the horizontal axis of the diagram represent the distance covered during each time increment Δt , and the mean speed V_m of each time interval is given by the values of V_{m1}, V_{m2} etc. Thus, within the first 5 seconds of starting from rest, the vehicle covers 23 ft and reaches a speed of 12 m.p.h., during the second 5 seconds it covers an additional 120 ft and attains a speed of just over 20 m.p.h., and so on. These values of Δt and Δl are indicated on a common straight line in Fig. 4, so that the distance covered at any time can be seen at a glance. In addition, the vehicle speed can also be plotted as a function of both distance and time. This gives all relevant information in one curve.

To determine acceleration up or down a gradient, the V axis in Fig. 3 is shifted respectively up or down. For example, the additional tractive resistance encountered on a grade of 1 in 20 is 112 lb/ton, and the axis is therefore moved to the position indicated by the dotted line in Fig. 3. The 5 sec triangles are again drawn with the 45 deg set-square. In this instance, a speed of 23.8 m.p.h. is reached in 60 sec. To verify the accuracy of this method the speed-time and speed-distance curves were also determined, for speed increments of 2 m.p.h., by the conventional analytical method. The results, plotted in Fig. 5, are in close agreement throughout. According to the graphical plot, the speed of 50 m.p.h. is reached after 75 sec, on covering a distance of 3,990 ft, and the results obtained by the analytical method are 75.83 sec and 4,012 ft respectively.

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3. W. GSCHUNG; "Das Voith-Diawabus-Getriebe", A.T.Z., Vol. 55 Nr. 3, March 1955.

INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

MARCH

Derby

Monday, 19th March, 7.15 p.m., in the Midland Hotel, Derby. Paper: "Automobile Design from the Safety Angle To-day and To-morrow", by A. Bourne.

North-Eastern

Wednesday, 21st March, 7.30 p.m., in the Chemistry Lecture Theatre, The University, Leeds. Paper: "The Size, Structure and Shape of European Automobiles", by Laurence Pomeroy.

North-Western

Tuesday, 20th March, 7.15 p.m., in the Engineers' Club, Manchester. Paper: "Disc Brakes", by F. G. Parnell, M.I.Mech.E., and F. J. Bradbury, A.M.I.Mech.E.

Scottish

Monday, 19th March, 7.30 p.m., in the Institution of Engineers and Shipbuilders, 39 Elmbank Crescent, Glasgow. Paper: "Front Suspension and Tyre Wear", by V. E. Gough, B.Sc.(Eng.), A.M.I.Mech.E., and G. R. Shearer, G.I.Mech.E.

APRIL

London

Tuesday, 17th April, 5.30 p.m., at the Institution of Mechanical Engineers, 1 Birdcage Walk, Westminster, S.W.1. Paper: "Front Suspension and Tyre Wear", by V. E. Gough, B.Sc.(Eng.), A.M.I.Mech.E., and G. R. Shearer, G.I.Mech.E.

North-Eastern

Wednesday, 11th April, 6.30 p.m., at the Cleveland Scientific and Technical Institute, Middlesbrough. Paper: "Rocket Propulsion", by Professor A. D. Baxter, M.Eng., M.I.Mech.E.

North-Western

Tuesday, 10th April, 7.15 p.m., in the Offices of Leyland Motors Ltd., Manchester. Paper: "Commercial Vehicle Structure", by P. Bruntton.

Representative examples of outer-seal and inner-seal Nilos rings for ball or roller bearings



THE NILOS RING

A Simple All-metal Sealing Device for Ball and Roller Bearings

EFFECTIVE sealing of anti-friction bearings to retain lubricant and to prevent the ingress of water, mud, dust or grit is often beset with difficulties owing to the limited space available and the complete lack of accessibility, in many instances, when the component is assembled on the vehicle. The Nilos sealing ring, manufactured by Ziller and Co., Achenbachstrasse 26, Dusseldorf, and marketed in Britain by Thomas Mercer Ltd., Eyewood Road, St. Albans, Herts., offers a possible solution of such problems. A seal of the single ring type in no instance exceeds the diameter of the bearing to which it is fitted and the axial width ranges from 1.5 to 5.0 mm respectively for the smallest to the largest of the rings produced. Individually, their weight is insignificant, the rings being pressed from strip metal 0.3 mm thick for overall diameters up to approximately 125 mm and from 0.5 mm strip for the larger diameters. The range of ring diameters is most extensive, progressing in small increments to suit ball and roller bearings of all commonly used standard sizes.

Rings are normally of zinc plated alloy steel but alternatively they can be supplied with a brass coating. For marine

use, or for other applications in salt-laden atmospheres, they can be produced in Tombak, a special phosphor bronze alloy. After pressing from the pre-plated strip material, each ring is individually set up and the sealing lip of the upstanding flange is turned to ensure correct axial height and squareness of edge.

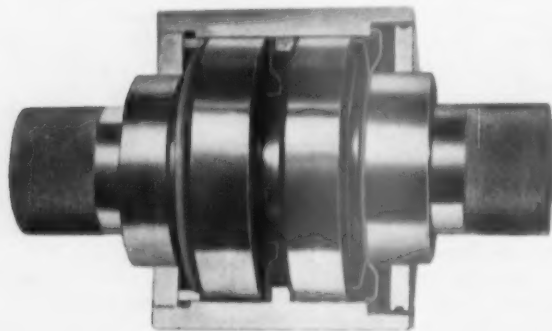
When mounted in position, with the seating annulus clamped against the side of one of the bearing rings, the sealing lip is resiliently constrained against the side face of the other bearing ring. In operation the sealing lip tends to wear a shallow groove in the bearing ring, the parts eventually forming a miniature, close-fitting labyrinth gland which prevents the escape of bearing grease and, conversely, the ingress of foreign matter. It is this feature that gives the ring the outstanding characteristic of becoming a more effective seal as its working life is extended.

Journal bearing applications

For journal bearings of either ball or roller type, alternative outer-seal and inner-seal rings, designated A.V. and J.V. types respectively are available. Functionally, the choice of type is not critical and is usually determined by considerations of bearing design and convenience of construction. The inner-seal type is to be preferred as the length of the sealing lip is minimized and the centrifugal force exerted by the lubricant at the sealing face is relatively reduced. Outer-seal rings are used mostly in applications where the shaft or axle is stationary and the outer ring of the bearing rotates. Where bearings must operate in exceptionally exposed positions or under arduous conditions of service, two or more Nilos rings may be mounted in series to afford complete protection.

A few commonsense precautions must be observed in mounting to ensure satisfactory operation. The diagrams, which are not drawn to scale, are included as typical examples of good practice. It is essential that the ring is fitted and supported accurately concentric with the bearing; the possibility of any departure from that position must be guarded against. It follows that an inner-seal ring must have a nominal diameter the same as the outside diameter of the

Sectioned model showing outer-seal and inner-seal rings on a journal bearing and a double-seal ring on a taper roller bearing



bearing and an outer-seal ring the same nominal diameter as the bore of the bearing. In no circumstances should the ring be positioned in way of a screw thread, a thread run-out, a thread or corner relief, or a free clearance. Such undesirable features may be avoided as indicated in diagrams A, E and F.

Rings must be securely clamped to obviate the possibility of slipping in operation. The shoulder height of housings or shafts must be adequate for this purpose or an intermediate ring, as provided at B in conjunction with a spring circlip, should be fitted. A very narrow shoulder can distort the edge of the ring and, to some extent, relieve the tension on the sealing lip.

A correctly designed and precisely machined bearing housing, in conjunction with an accurate shaft assembly, will automatically ensure the appropriate stressing of the ring and eliminate the need for any checking, adjustment or selection when fitting. The sealing lip should not engage the manufacturer's name and identification numbers, usually stamped on the outer bearing ring, or sealing may be impaired. With unilateral sealing this can be arranged by fitting the bearing with the marked face on the opposite side to the seal and when bilateral sealing is employed a J.V. inner-seal ring is used on the marked side. Should this not be convenient the marked side of the bearing should be arranged to face the inside of the component, where there is less risk of foreign matter entering. In instances where the interruption of the sealing face by this marking cannot be avoided, a special ring which is more highly stressed than the standard type can be supplied. This accelerates the formation of the sealing groove which it cuts through to a depth below that of the stamp impression.

Axial or angular movement of the shaft would result in the impairment or even the complete loss of sealing effect. Consequently, these rings should be fitted to bearings of the self-aligning type only when the bearings are used on account of their greater load-carrying capacity and are run in their neutral position with no possibility of making self-aligning movements. Similar considerations govern the fitting of the sealing rings to cylindrical roller bearings of the type permitting an axial float in one direction, as in diagram C. In such cases the shaft must be positively located so that no axial movement can occur.

The two diagrams C and G of cylindrical roller bearing applications also show the rings arranged to seal on housing faces instead of on the bearing races. In both of these instances the bearing housing is drilled to enable the bearing to be replenished with grease without the necessity to dismantle the sealing rings. In all cases, however, the annular depression in each ring is filled with suitable bearing grease before it is mounted in position.

Two standard rings, of either the inner-seal or outer-seal type, may be spaced by an intermediate ring to seal respectively on the housing and the bearing. The intervening cavity is filled with a water-resistant grease in order to render the bearing positively waterproof. Diagram D shows such an arrangement, while in diagram F a special reversed-flange ring obviates the need for an intermediate ring.

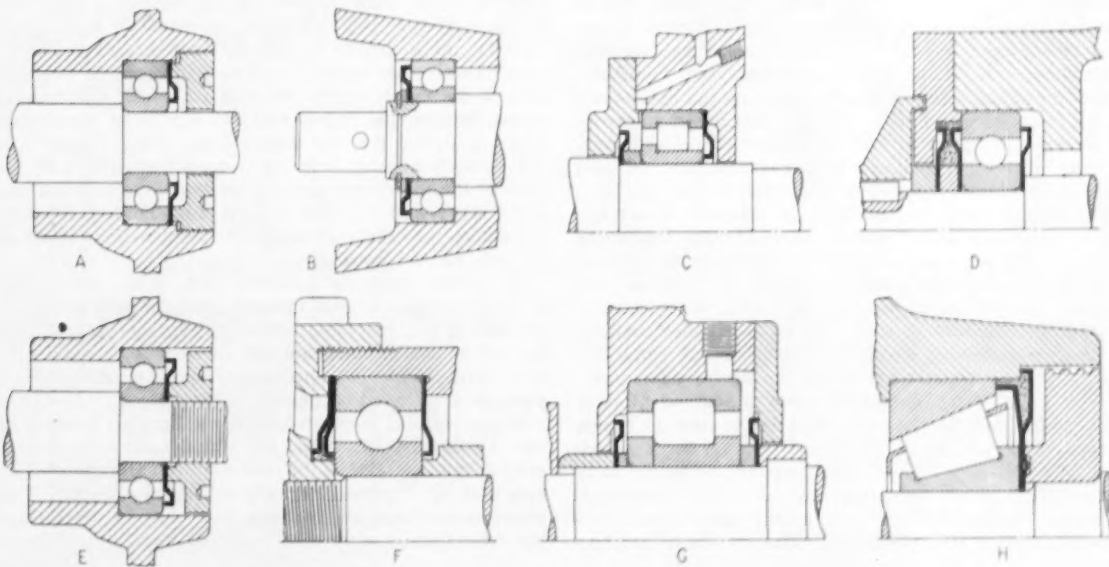
Sealing taper roller bearings

Modified types of sealing rings are available to meet the special requirements of taper roller bearings. A single-seal ring is of the A.V. type but of greater axial width than the standard versions in order to pick up the offset faces of inner and outer bearing rings. The double-seal arrangement comprises an A.V. type ring supplemented by a special ring, termed the A.K. type, to seal on the housing, as in diagram H showing a wheel hub mounting. As in the previous double-sealing assemblies, the space between the rings is loaded with a water-resistant grease. To prevent the possibility of slip occurring, the A.K. ring is formed with two dimples spaced at 180 deg. These are accommodated either in a pair of holes drilled in the face of the clamping or spacing collar or, alternatively in an annular groove fitted with a small driving pin.

Present availability is for rings suitable for ball or roller journal bearings of from 3 mm to 200 mm bore and from 10 mm to 360 mm outside diameter. In the case of taper roller bearings, standardized sizes range from 15 mm to 150 mm bore and from 40 mm to 270 mm outside diameter. Nilos rings of all designs are protected by patents.

In addition to the obvious applications for motor vehicle components, the Nilos ring can be used on production equipment to seal bearings against the ingress of workshop dirt. They are functioning satisfactorily on electric motors, machine tools, conveyors and other plant.

Typical methods of mounting Nilos rings: A—inner bearing seal; B—outer bearing seal, retained by circlip; C—inner bearing and outer casing seal, replenishable; D—double outer rings, grease sealed; E—outer bearing seal; F—double inner rings, grease sealed; G—bilateral outer casing seals, replenishable; H—double rings for taper roller bearings, grease sealed



LLOYD AIR-COOLED ENGINE

A 596 cm³, Two-Cylinder, Four-Stroke Unit With an Overhead

Camshaft and Hemispherical Heads

A NEW air-cooled engine has recently been introduced by Lloyd Motoren Werke G.m.b.H. It is a two-cylinder, four-stroke unit with a bore and stroke of 77 mm and 64 mm respectively, a swept volume of 596 cm³, and a compression ratio of 6.6:1. The unit develops 19 b.h.p. at 4,500 r.p.m. Thus, the output in terms of b.h.p./litre is 31.6. That it is not higher than this, despite the employment of an efficient combustion chamber and valve gear arrangement, is no doubt due to the relatively low compression ratio.

There are, of course, a number of fundamental differences between the requirements for air-cooled engines for cars and for motor cycles. The main differences are in the shape of the space available for engine accommodation, the cooling arrangements, and the transmission requirements. Although it is possible to use a horizontally-opposed cylinder layout and to place it just inside a suitable opening in front of the bonnet, as in the C.E.M.E.C. engine illustrated in the November 1953 issue of *Automobile Engineer*, this arrangement is more suited to side valve than overhead valve engines. Accordingly, Lloyd have adopted the twin-cylinder, in-line layout, which enables them to incorporate a compact, simple and accessible valve gear and at the same time to employ a cylinder head of approximately hemispherical form.

The penalty for arranging the cylinders in this way is that a ducted blower has to be employed for cooling the unit. This blower is mounted in tandem with the generator, on the side of the crankcase, and is belt-driven from a pulley on the front end of the crankshaft. In order that it shall not overhang excessively to one side, the engine is inclined so that the cylinders overhang slightly to the left and the blower slightly to the right of the crankcase; the inlet port of the cylinder head extends above the cooling air blower, and the downdraught carburettor is mounted on top of it.

Although both the cast iron cylinders and aluminium cylinder heads are separate components, the rocker box mounted above them is in one piece. Six nuts on short studs screwed into the crankcase hold down the cylinders by their flanges at their lower ends. The centre pair of nuts help to hold down the flanges of both cylinders. Location of the cylinders is, of course, effected by spigoting their lower ends into the holes in the crankcase. Similarly, the cylinder heads are located by spigoting the upper ends of the cylinders into them. The cast aluminium crankcase is divided, on a plane normal to the axes of the cylinders, at the level of the centre of the crankshaft. A cast aluminium sump is employed; it incorporates a transverse web, approximately midway between its ends, to support the intermediate main journal bearing. An interesting feature is that the joint washer, which is housed in a groove in the joint face of the sump, is only employed in the joint on one side of the engine, that is, on the side which is lowest when the unit is canted over in its installed position.

The three-bearing crankshaft is fabricated from five pieces and is symmetrical about its centre component, except in that the rear end extension is adapted to carry the flywheel and the front end extension to take the timing drive. On each side of the centre bearing, a crank web and pin, which are integral, are assembled on to the main journal; the other two components of the crankshaft assembly are the front and rear webs, together with their integral journals and extensions. This fabricated arrange-

ment has been adopted so that the ball and roller bearings employed for the main journals can be assembled on to the shaft. It would seem to be difficult to justify, on the grounds of compactness, the employment of rolling element bearings, since there appears to be plenty of room for plain bearings, except possibly at the rear. However, it is likely that the high coefficient of expansion of the aluminium alloy housings and the large surface area of the joint faces between the halves would present problems, so far as the accommodation of shell type bearings is concerned.

The crankshaft in this form can be manufactured with relatively simple machine tools. With this end in view, the flywheel has been keyed on to the rear extension of the shaft, which, therefore, is of plain cylindrical section, and the inner race of the ball bearing of the rear main journal is clamped between the flywheel and a shoulder on the shaft. This whole assembly is retained by a single nut on a waisted stud in an axial hole drilled through the extension. The tapping for the stud is in the crank web; doubtless this obviates trouble due to stress concentrations. The length of the stud ensures adequate resilience to accommodate differential rates of thermal expansion of the assembly. Axial location is effected at this rear bearing, the outer race being located between a snap ring and a plain washer in grooves in its housing. This washer acts as an oil-baffle at the outer end of the bearing.

Both the other main journal bearings are of the roller type. The outer race of the intermediate one is located in the same way as that of the rear bearing. Its inner race, together with the gear that drives the oil pump, also forms a distance piece that separates the adjacent crank webs. The gear is dowelled to the foremost of these two webs. Assembled on to the front end of the crankshaft in the following order are: a distance sleeve, the roller journal bearing, a second distance sleeve, the timing drive sprocket and the keyed-on pulley. The whole assembly is retained by a nut on the front end of the shaft.

An oil-baffle washer is fitted in the front end of the housing for the outer race of the bearing, which is located at the rear end by a short flanged sleeve, of light gauge material, in a groove round the housing. This sleeve extends rearwards into a shroud ring mounted on the front face of the crank web. Oil splashed out of the bearing passes through the sleeve, and is collected in the shroud ring and passed into the hollow crank pin. Thence it is fed through a radial hole into an annular groove in the bore of the inner race of the roller type, big end bearing. From this groove it passes through radial holes to lubricate the rollers. A similar arrangement is employed to lubricate the rear big end bearing.

Oil supply from the pump to each of the three main journal bearings is taken through a longitudinal gallery in the base of the sump, and vertical passages in the front and rear walls, and intermediate web in the sump casting. An interesting feature of this arrangement is that the oil passages are formed by pipes cast in the sump. The front and rear vertical passages and the gallery are formed by one continuous pipe, while the vertical one in the intermediate wall, as well as the feed from the oil pump, which also is in the intermediate wall, are separate pieces. This arrangement eliminates drilling and plugging operations, but, nevertheless, is not easy from the foundryman's point

of view. The oil pump, which is driven by the spur gear between the front of the intermediate bearing and the adjacent crank web, is mounted on the transverse web in the crankcase. All the camshaft and rocker bearings are lubricated by splash, the oil originally entering the rocker box from a trough above the front bearing of the camshaft. Scavenge oil returns through the timing drive housing.

A relatively long chain drives the camshaft. Its lower end is housed in chambers formed in the crankcase and sump castings, while its upper end is enclosed by a forward extension of the rocker box. A sleeve connecting the two parts of the housing completes the enclosure of the chain drive. This sleeve is flanged at its lower end, where it is bolted to a machined face on top of the front end of the crankcase. A spigot beneath the forward portion of the rocker box projects into the upper end of the sleeve. The seal at the top end is effected by a rectangular section rubber ring, and a conventional joint washer is used at the lower end. A cast aluminium cover is spigoted to the front end of the rocker box to give access to the half speed wheel.

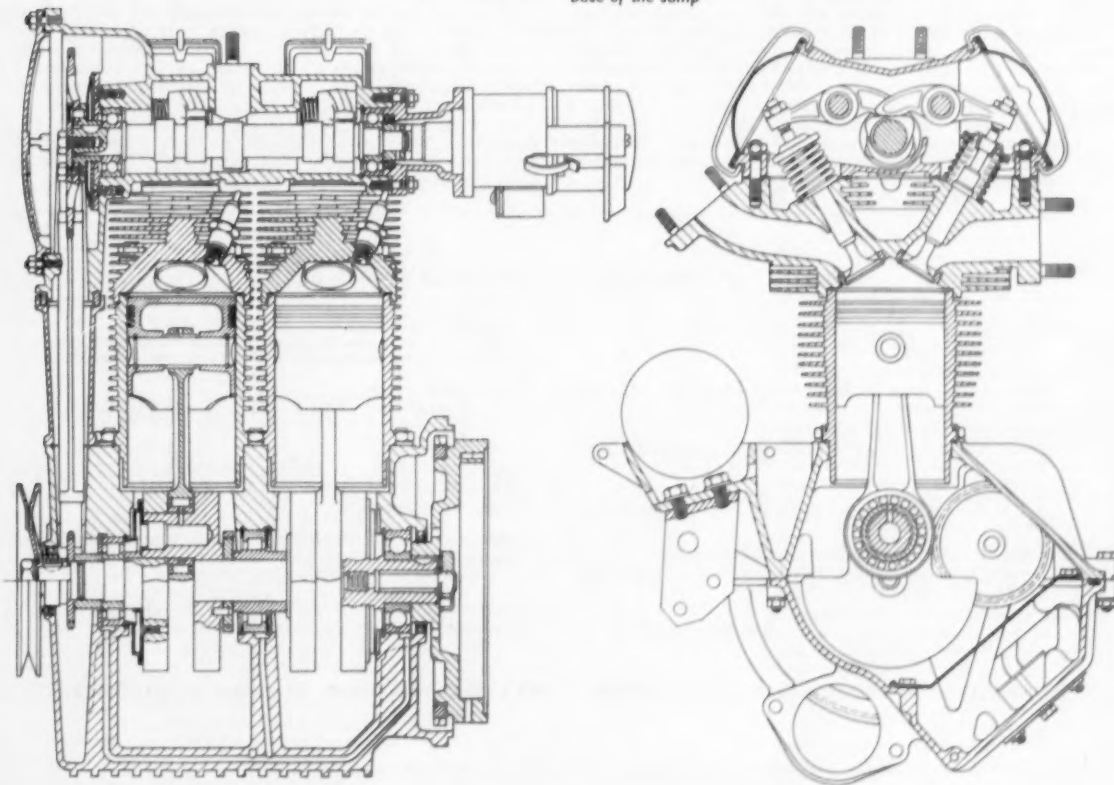
The half speed wheel is a dished pressing, of thick section, spigoted on to a cast iron hub, which in turn is keyed-on to the end of the camshaft. This whole assembly is pulled against the inner race of the front ball bearing by a retainer plate, together with a set bolt screwed axially into the front end of the camshaft. Bolts round the periphery of the retainer plate are passed through clearance holes in the half speed wheel and screwed into the flange of the hub to prevent relative rotation between the components of the assembly. The outer race of the ball bearing is allowed a limited amount of axial float in its housing and is retained by a cover spigoted into the front end of the housing and secured by countersunk set screws. On the front face of this cover is a thin gauge plate; a Belleville

washer to pre-load the ball bearing at the rear end of the camshaft, where axial location is effected, is interposed between it and the half speed wheel. The outer race of this bearing is clamped between a snap ring, in a groove in the housing, and the spigot end of a casting that carries the contact breaker and distributor, which is secured to the rear face of the rocker box. Between the pairs of cams for each cylinder is an eccentric for actuating the fuel pump. This pump is mounted on top of the rocker box.

There are two rocker shafts, one on each side of the camshaft. They are supported at their ends in the front and rear walls of the rocker box, and at their centre by a boss, also cast in the rocker box. Coil springs are fitted round the shafts to hold the rockers against the bosses that carry the assembly. One end of each rocker arm bears directly on the camshaft and the other carries the tappet adjusting screw, which bears on the end of the valve. Because of the light weight of the reciprocating units of the valve gear, relatively light coil springs are employed. They are each retained by a simple pressed steel washer and split tapered collets.

Two small dished pressings, one on each side, form the rocker covers, through which access is gained for tappet adjustment and other servicing operations. An unconventional feature is that, together with their joint washers, these covers are each held down by two C-shaped wire clips, one at each end. The ends of the arms of each of the clips are carried in two blind holes, one above and the other below the aperture closed by the cover. For assembly, the spring clips are pivoted about the ends of their arms until they are clear of the aperture, so that the cover, together with its joint washer, can be put in place; then the clips are swung back and sprung into grooves near the ends of the crown of the cover.

In the Lloyd air-cooled engine a fabricated crankshaft is employed, and the oil gallery and passages leading from it are tubes cast in the base of the sump



Recent Publications

Brief Reviews of Current Technical Books

Berechnung und Gestaltung von Gummifedern. (The calculation and Design of Rubber Springs)

By E. F. Göbel, in German.

Berlin: SPRINGER-VERLAG, Germany. 1955. Second Edition.
6½ x 9. 86 pp. Price 9 D.M.

This book is the seventh of a series comprising sixteen volumes dealing with the design of machine components. The series is edited by Prof. K. Kollmann, a distinguished engineer well qualified in the design and development of road vehicles in general and their engines in particular. In common with two previously reviewed volumes of this series, this work is of great value to automobile engineers, since it contains much valuable information presented in a clear, lucid and directly useful manner.

The first chapter occupies only five pages and serves as an introduction to the applications and manufacture of rubber components. This is followed by another on the elastic and vibration damping characteristics of rubber elements, their noise suppressing properties, strength, and the effect of high and low temperatures on their elasticity and endurance. Dimensional requirements of rubber elements, as calculated on a static load basis, are considered next; because of its importance, this subject is dealt with comprehensively and, in fact, in greater detail than in any earlier publication on this subject. The elements thus dealt with are: the concentric cylindrical type stressed in shear, both as purely cylindrical as well as conical units; cylindrical and double conical units stressed in shear; disc units in torsion; simple rubber elements in compression, as well as subjected to combined compressive and shear stresses. Transverse stressing of concentric cylindrical elements and combined stressing of such units are also discussed. The principal equations obtained from this analysis are summarized in a table at the end of the chapter.

Calculations for anti-vibration elements subjected to dynamic loads are considered next. Then, after an introduction into the fundamentals of vibration mechanics, the author deals in great detail with the design of anti-vibration mountings for foundry sieves operated by rotary unbalance units. This, as well as the previous chapter, will be of particular interest to automobile engineers, because in a more generalized form these problems and their methods of solution are also encountered with vehicles.

Design examples are given in the fifth chapter. In this, the design of mountings for compressors, pumps, ventilators, engines, sewing machines, machine tools, road and rail vehicle suspensions, shaft couplings, etc. is discussed. Methods of testing rubber springs are briefly considered in the last chapter. Finally, a bibliography comprising 120 references is included. The author has succeeded in presenting a great deal of valuable information in an authoritative form of immediate use to designers; thus, he has satisfied a long-felt need.

The Testing of High Speed Internal Combustion Engines

By Arthur W. Judge, A.R.C.Sc., D.I.C., Wh.Sc.,
A.M.I.Mech.E., A.M.I.A.E.

London: CHAPMAN AND HALL LTD., 37 Essex Street, W.C.2.
1955. 9½ x 6. 494 pp. Price 75s.

In recent years, there have been notable developments in engine testing methods and equipment, and in the fourth edition of this well-known book, much new information and data have been included. The section dealing with the methods of measurement of horse power has been rewritten and expanded. It now includes accounts of recently developed brakes and dynamometers with special reference to electrical and power recuperation units; modern eddy current and dynamic units have also received special consideration. In view of the ever-increasing importance of developments in aircraft gas turbine design and production, an entirely new section devoted to the subject of testing this type of gas turbine has been added.

Plastics Progress 1955

Papers and Discussions at the British Plastics Convention 1955
Edited by Philip Morgan, M.A.

London: ILIFFE & SONS LTD., Dorset House, Stamford Street,
S.E.1. 1956. 9½ x 6. 432 pp. Price 50s.

As the technology of plastics is extending so rapidly in so many diverse directions, some difficulty is experienced in keeping abreast of new developments. Each year a convention is organized, in connection with the British Plastics Exhibition, at which specialists in their respective fields read papers on various aspects. In this volume the full texts of the papers presented at the 1955 Convention are given, together with a complete transcript of the ensuing discussions. The informal character of the discussions, as recorded, is retained.

Papers are grouped into chapters under nine heads: Polymer structure and properties; Expanded plastics; Thermoplastics; Extrusion; Work study and productivity; Injection moulding; Patents; Foundry resins; and Glass-reinforced plastics.

Of special interest to automobile engineers are the papers "Developments in the use of plastics in the foundry" (P. G. Pentz, Leicester, Lovell & Co. Ltd.), "Consistency in glass fibre-reinforced moulding" (J. Rees, Bristol Aeroplane Co. Ltd.), "Glass-reinforced plastics in automobile construction" (H. Silman, Ford Motor Co. Ltd.), "The testing and development of high quality glass-fibre/polyester laminates" (P. H. H. Bishop, R.A.E. Farnborough and E. Haythornwaite, Marglas Ltd.) and "Epoxide resin/glass laminates" (W. J. Marmion, Shell Chemicals Ltd.).

BOOKS of interest to AUTOMOBILE ENGINEERS

AUTOMOBILE CHASSIS DESIGN

By R. Dean-Avers. 2nd Ed. 30s. net. By post 31s. 1d.

AUTOMOBILE EFFICIENCY

Maintaining Performance with Electrical Test Equipment. By E. T. Lawson Helme, A.M.A.E.T., A.M.I.M.I. 10s. 6d. net. By post 11s. 1d.

AUTOMOBILE ELECTRICAL EQUIPMENT

By A. P. Young, O.B.E., M.I.E.E., M.I.Mech.E. and L. Griffiths, M.I.Mech.E., A.M.I.E.E. 5th Ed. 25s. net. By post 26s. 1d.

DIESEL MAINTENANCE

By T. H. Parkinson, M.I.Mech.E. Edited by Donald H. Smith, M.I.Mech.E. 4th Ed. 12s. 6d. net. By post 13s. 4d.

ELECTRICAL SERVICING OF THE MOTOR VEHICLE

Principles, Design and Choice of Test Apparatus. By E. T. Lawson Helme A.M.A.E.T., A.M.I.M.I. 8s. 6d. net. By post 9s. 2d.

GAS TURBINES AND JET PROPULSION

By G. Geoffrey Smith, M.B.E., Revised by F. C. Sheffield. 6th Ed. 35s. net. By post 36s. 5d.

THE MODERN DIESEL

Automotive Oil Engines for Transport, Marine and Industrial Use. By Donald H. Smith, M.I.Mech.E. 12th Ed. 12s. 6d. net. By post 13s. 6d.

THE MOTOR VEHICLE

By K. Newton, M.C., B.Sc., A.C.G.I., A.M.Inst.C.E., M.E.Mech.E., and W. Steeds, O.B.E., B.Sc., A.C.G.I., M.I.Mech.E. 5th Ed. 35s. net. By post 36s. 5d.

SERVICING GUIDE TO BRITISH MOTOR VEHICLES, CARS, COMMERCIAL VEHICLES AND TRACTORS. Vol. 1.
63s. net. By post 65s. 6d.

SERVICING GUIDE TO BRITISH MOTOR VEHICLES, CARS, COMMERCIAL VEHICLES AND TRACTORS. Vol. 2.
42s. net. By post 44s.

SERVICING GUIDE TO BRITISH MOTOR VEHICLES, CARS, COMMERCIAL VEHICLES AND TRACTORS. Vol. 3.
50s. net. By post 52s.

Obtainable at all booksellers or from: ILIFFE & SONS LTD Dorset House, Stamford St. London, S.E.1.

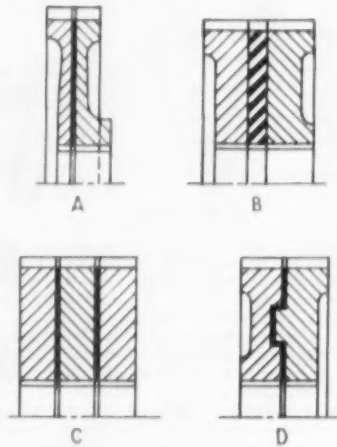
CURRENT PATENTS

A REVIEW OF RECENT AUTOMOBILE SPECIFICATIONS

Cam and roller steering gear

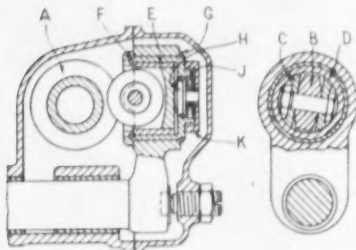
THE engaging roller of this continuous cam type steering gear is arranged to rotate on its axis, to be rotatable about an axis parallel to the rocking axis, and to be displaceable along the second axis against the constraint of a spring. In general arrangement the gear is conventional with the steering worm A mounted on ball bearings drawn up by a screwed sleeve which is secured in its adjusted position by a pinch bolt. The rocker shaft runs in plain bearings and is held up by a screwed adjuster engaging a hardened thrust pad.

Novelty lies in the roller assembly carried in the head of the rocker shaft arm. This comprises a roller B running on a spindle bolted across the limbs of a forked carrier C which is pressed into a cylindrical jacket D of a hard, wear-resistant material. The jacket is rotatably and slidably accommodated in a bearing bush E, being retained with predetermined axial clearance by circlip F. A circlip G secures the bush in the rocker arm. On the side remote from the point of engagement, the roller assembly is supported by



No. 730332

non-metallic layer is uniform throughout the area. A wider-faced gear may have a thick layer of plastics material, as at B, or be formed of more than two laminations, as at C. To increase the bonding area, the metal parts of gear D are formed with inter-engaging annular configurations. Patent No. 730332. Willys Motors, Inc. (U.S.A.)



No. 728923

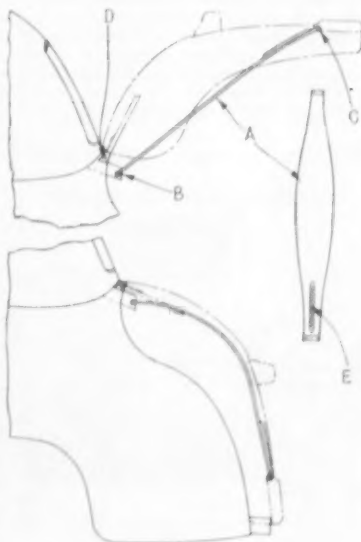
a pad H, shouldered to limit its range of movement and loaded by plate springs J retained by a screwed cap K. Under heavy shocks transmitted from the road wheels the roller assembly is displaced and cushioned by the energy stored in the plate springs.

Towards the ends of the arc described by the end of the rocker arm, the roller tends to move out of engagement with the cam track and play will be increased. To correct this, the worm is given an hour-glass formation by reducing the depth of the thread, and consequently the width of the track, from the centre towards each of the ends. Patent No. 728923. Zahnrad-fabrik Friedrichshafen A.G. (Germany).

Silent-running gear

BY forming a timing gear of two or more metal laminations bonded together by a layer or layers of a plastics cement, the resulting unitary component is substantially non-resonant. It has virtually the same strength and wearing qualities as a normal metallic gear. A thermosetting material is used for the bonding layer but a thermoplastic material may be employed if the gear is to be operated at comparatively low temperatures.

Several methods of construction are suggested. At A the two metal portions are machined or ground on the inner faces so that the space occupied by the

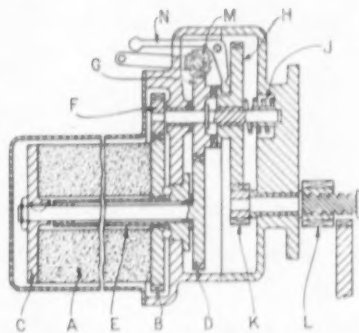


No. 729593

pins B and C bracketed respectively to the vehicle body and the boot lid. Pin B is arranged below a line joining the hinge axis D and pin C.

When the lid is open the strut is almost straight and effort is required to lower the lid to the closed position. The lifting force of the spring decreases as the lid is lowered and, at about two-thirds of its movement, becomes equal to the weight of the lid. Thereafter, the lid falls gently to the closed position. In that position the spring fits snugly into the concave contour of the lid and does not encroach on the luggage space. To conduce to this end, the lower portion of the spring may be formed with a rib E to resist local bending.

Specifically the form of the strut is not limited. It may be a single-leaf spring, a laminated spring, a spring rod, a bundle of rods clipped at intervals and moulded in rubber or plastics, or coiled or waved wires or strips. Patent No. 729593. Humber Ltd.



No. 728689

Torsional rubber engine starter

FOR agricultural tractors and other analogous applications where the conventional electric motor is not convenient or not desirable, an energy-storing starter is proposed. In the example the elastic component is a cylinder of rubber, but a metal spring or springs may be used.

To one end of the rubber cylinder A is bonded gear wheel B, and to the other end is bonded a flange C secured to a shaft carrying gear wheel D. This shaft is mounted in bearings in a fixed tubular member E extending through the bore of the rubber cylinder. The two wheels are connected by a layshaft carrying a fixed pinion F and a rotatable pinion G which can be clutched to the layshaft by face dogs on a slidable gear wheel H. Spring J holds the face dogs in engagement and wheel H engages pinion K on an output shaft carrying the starter pinion L.

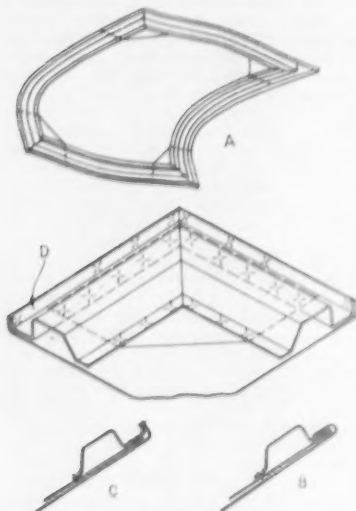
Rotation of worm M by means of a hand crank imparts angular movement at different rates to the ends of the rubber cylinder and produces in it a torsional strain. Simultaneously, the starter pinion L is engaged with the ring gear and the engine crankshaft is slowly turned over. On release of the clutch by lever N the energy stored in the rubber cylinder is transmitted to the output shaft to effect a start. Patent No. 728689. C.A.V. Ltd.

Panel reinforcement

IT is desirable that metal panels constituting such items as doors, bonnet lids and boot lids be torsionally stiff. Commonly they are reinforced by channel section members, one flange only of which is spot welded to the margin in order to avoid marking the outer surface of the panel. The reinforcing members are themselves mitred at the corners and welded.

Torsional stiffness is enhanced, according to this invention, by securing a gusset plate to each corner of the reinforcing structure. The gusset plate overlaps the flanges of the reinforcing members and provides either a straight or curved web at the corner joint.

Such a reinforcement is shown at A on the underside of a boot lid. In that instance the reinforcing structure is secured to the panel by folding the panel margin over the reinforcing channel flange and clinching, as shown in section B. An alternative method is to turn in the channel flange to lap an intumed marginal flange on the panel and spot welding together, as



No. 729772

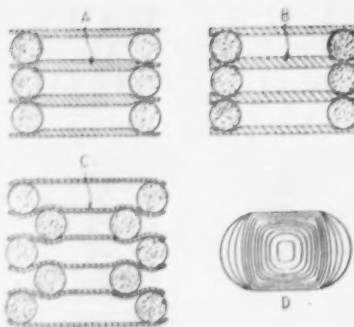
at C. These united flanges then constitute, with one wall of the channel reinforcement, an inwardly facing channel D which can serve as a drain, or receive a rubber sealing strip, or accommodate a tacking strip for interior trim. Patent No. 729772. Pressed Steel Company Ltd.

Rubber springs

SUSPENSION springs comprising a stack of rubber rings with metal separators have long been known and widely used on the railways. The rubber rings were commonly of a substantially rectangular cross section but an essential feature of this spring for motor vehicles is a ring of circular cross section.

Three examples are illustrated; the number of rings will, of course, be varied to meet requirements as regards load-supporting capacity and performance. In one, location is secured by means of shouldered metal parts A, while in another the metal parts B are formed with annular grooves to receive the rubber rings. The third type has rings of two different diameters spaced by pressed metal parts C.

As compared to rings of rectangular section, the following advantages are claimed:



No. 729022

(1) The compression load-deformation diagram of the spring can be varied between wide limits by modifying the cross-sectional form of the metal parts.

(2) Under compression, the deformation of the ring is such that resistance to stress in the ring is at an optimum value. Local deformations of the external surface, where fractures commence under crushing loads, are much reduced. Furthermore, deformations in the interior of the rubber mass are distributed in a regular manner without points of local concentration, as indicated at D.

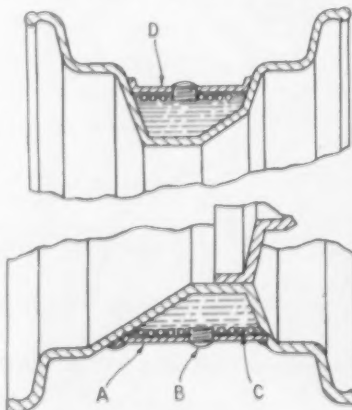
(3) The compression load-deformation diagram is of curvilinear form. Stiffness increases rapidly above a certain loading.

(4) Under dynamic strain a certain amount of relative sliding movement occurs, involving a dispersion of energy, giving an inherent damping effect. Patent No. 729022. Societa Applicazioni Gomma Antivibranti S.A. (Italy).

Automatically balanced wheels

AN out-of-balance wheel not only causes uneven tyre wear but is also liable to increase bearing wear. The proposed method of balancing is claimed to be automatically self-adjusting while the wheel is rotating.

The sides of the well in the rim are provided with shallow grooves internally to receive a metal band A which is welded to the rim, and at its abutting ends, to form a closed annulus. Through an orifice closed by a screwed plug B a plurality of spherical elements C is introduced, and also a fluid to fill the annulus. The fluid should be of the type having a viscosity that is not appreciably affected

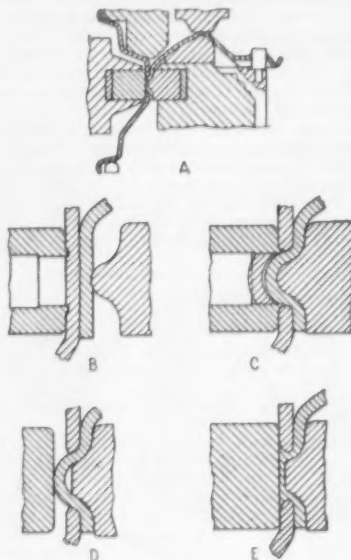


No. 730291

by temperature variations, and ethylene glycol is suggested. Steel or lead shot may be used as the spherical elements.

When the wheel is stationary the spherical weights fall to the lower run of the annulus. As the wheel is rotated the weights are flung outwardly by centrifugal force to the band A which is concentric with the axis of rotation. In a case of imbalance, when the heavy sector of the tyre strikes the road surface the force exerted will disturb the weights, which will float towards the opposite side of the wheel and thus reduce the impact when the heavy sector next strikes the road.

In an alternative construction the annulus may be formed by two complementary, semi-circular channels D welded together and into the rim. Patent No. 730291. J. C. Wilborn (U.S.A.).



No. 729864

Riveted wheel structure

THIS method of securing together a wheel disc and rim obviates the need for the separate punching of holes in the two parts and the feeding and insertion of rivets, thus simplifying and cheapening the assembly. The flanged disc is forced-fitted into the rim and a plurality of punch and die sets simultaneously raise protrusions on the disc flange which punch out slugs from the rim. In a follow-on operation, the heads of the protrusions, which extend through the rim, are upset to form, in effect, flush rivet heads in the holes.

The layout of the supporting fixture for the inter-fitted disc and rim is shown in the section A through one of the tool sets. In the first operation the profiled end of the hollow die is pressed into the rim, as at B, to form an annular depression that facilitates the subsequent development of the slug. The rim is then supported by the die, and the protruding end of the punch presses the disc metal through the rim to force out the slug, as at C.

A finishing die set is shown at D with the suitably shaped, flat-topped punch entered to support the disc metal, and at E where the flat-faced die has advanced to upset the protrusion. Patent No. 729864. The Budd Company (U.S.A.).

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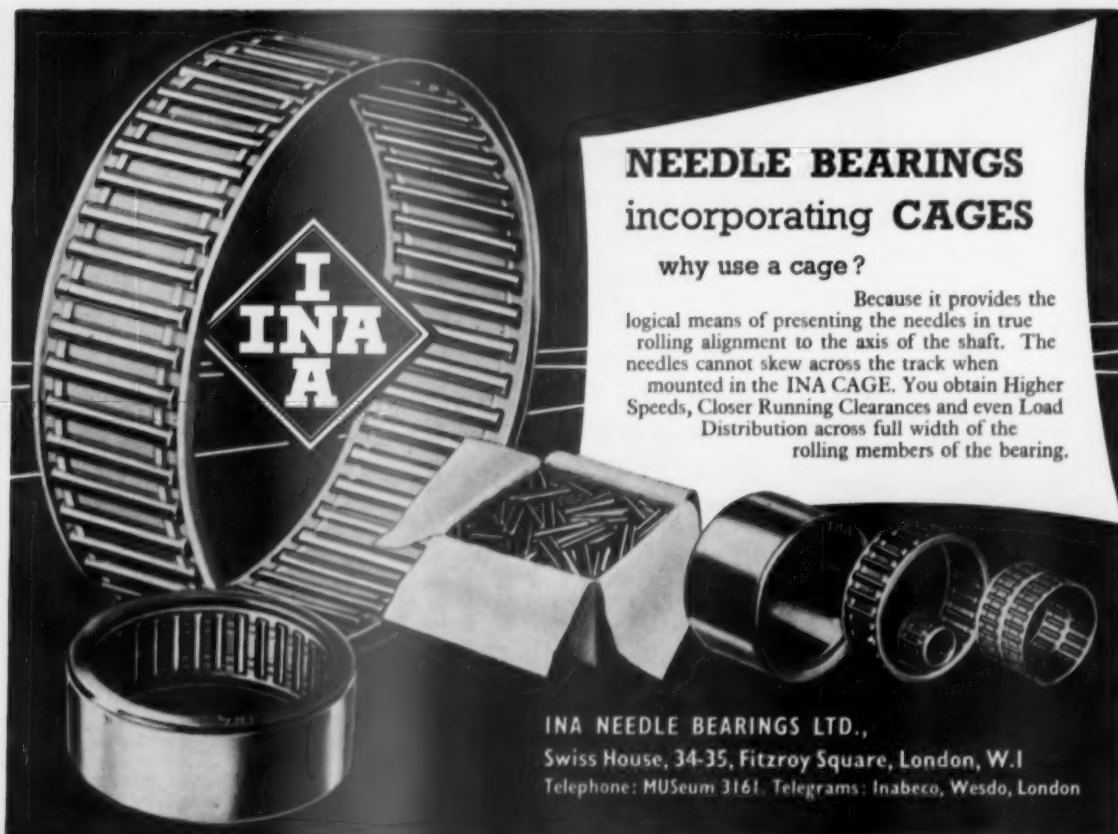
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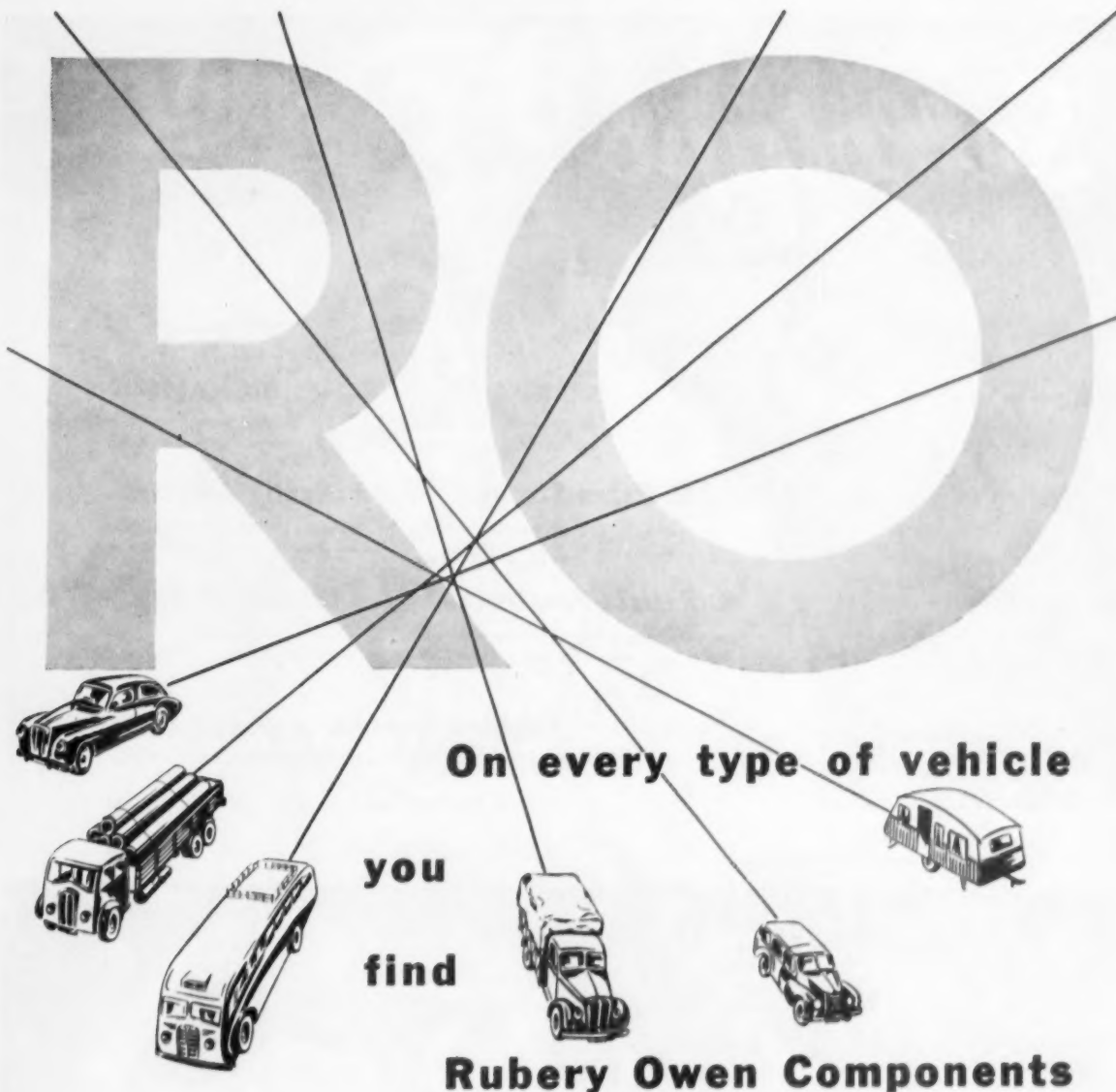


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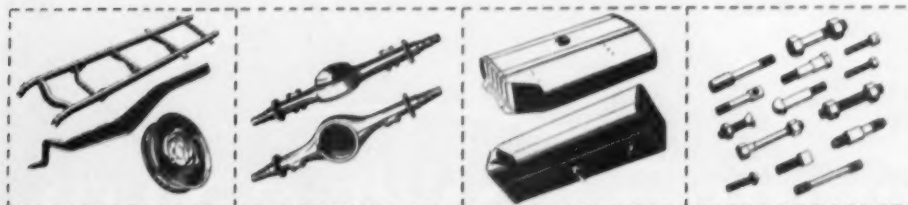
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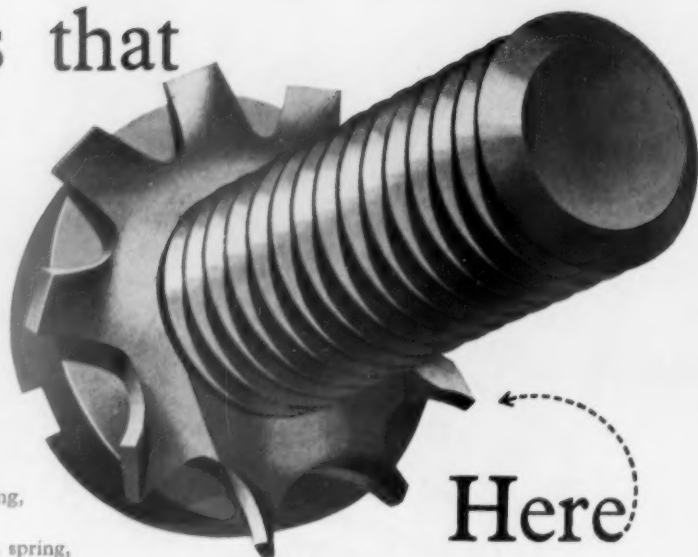
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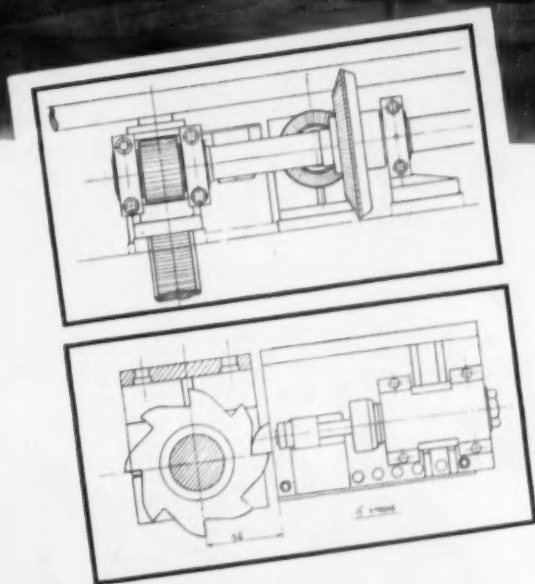
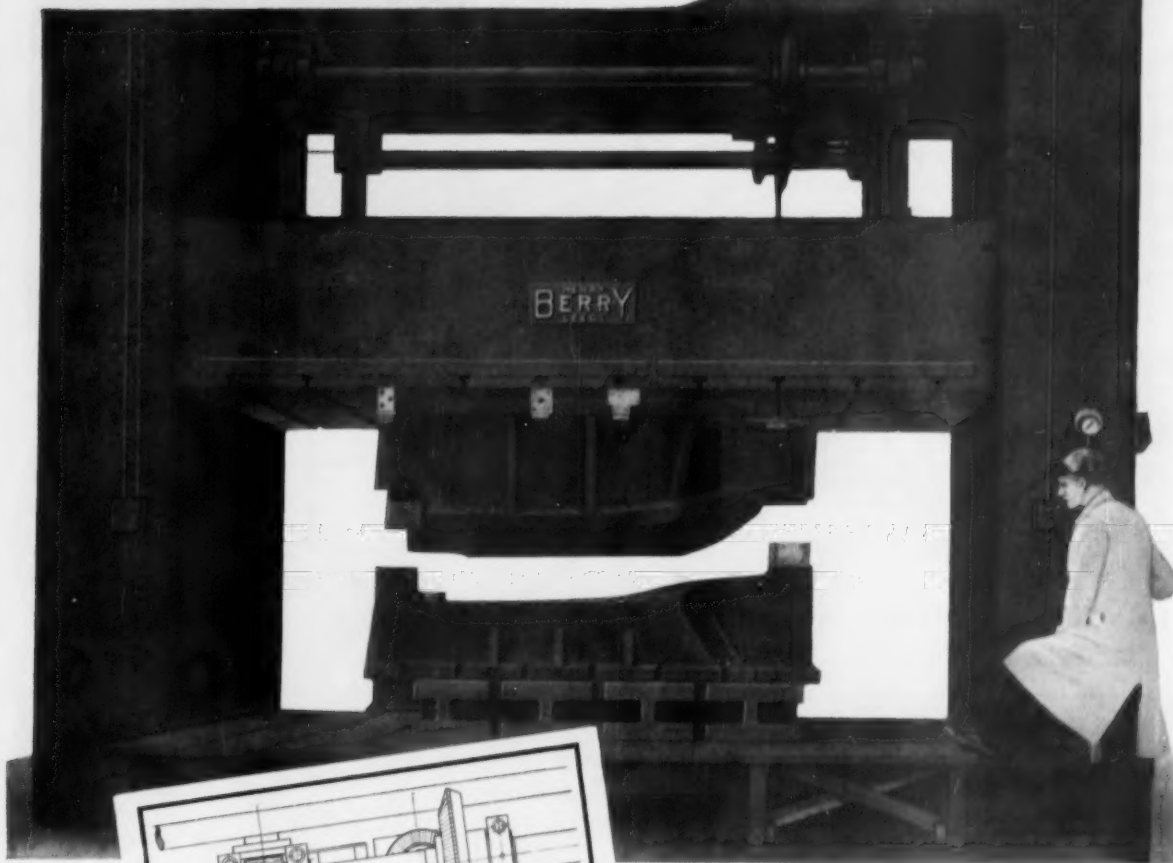
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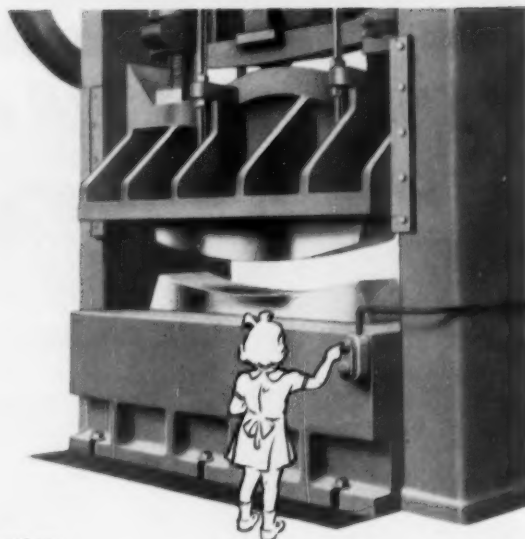
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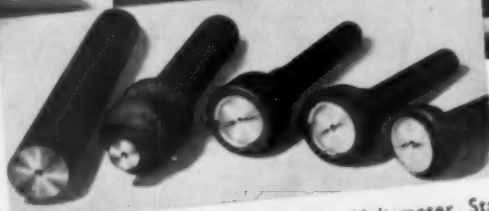
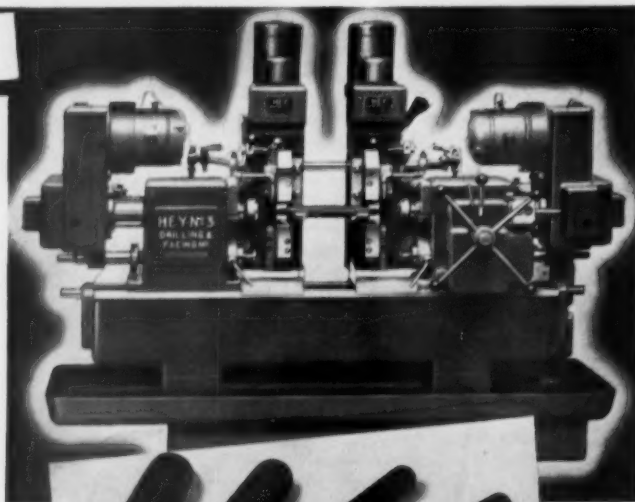
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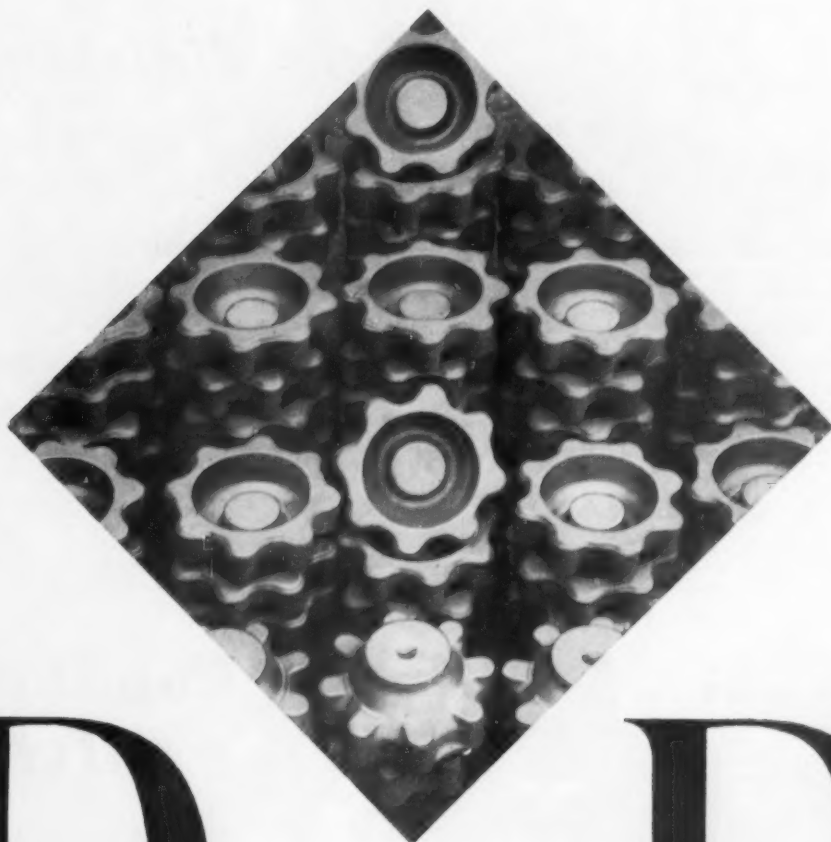
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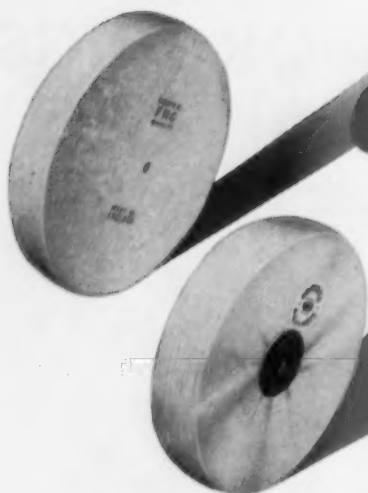


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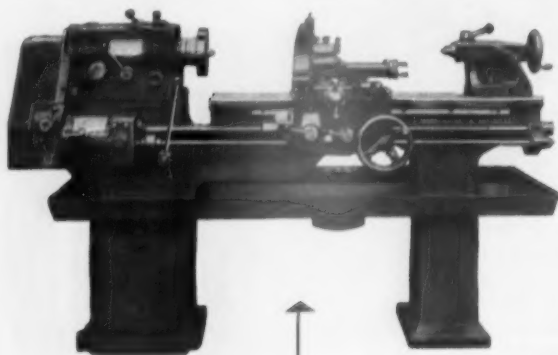
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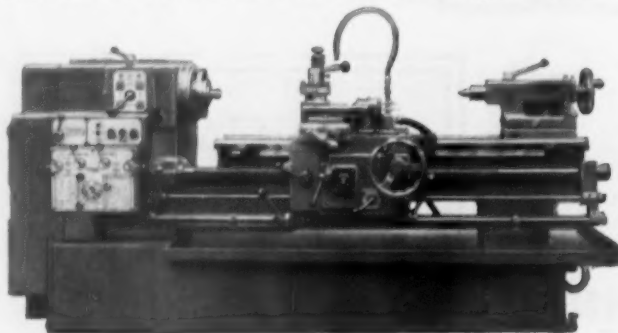
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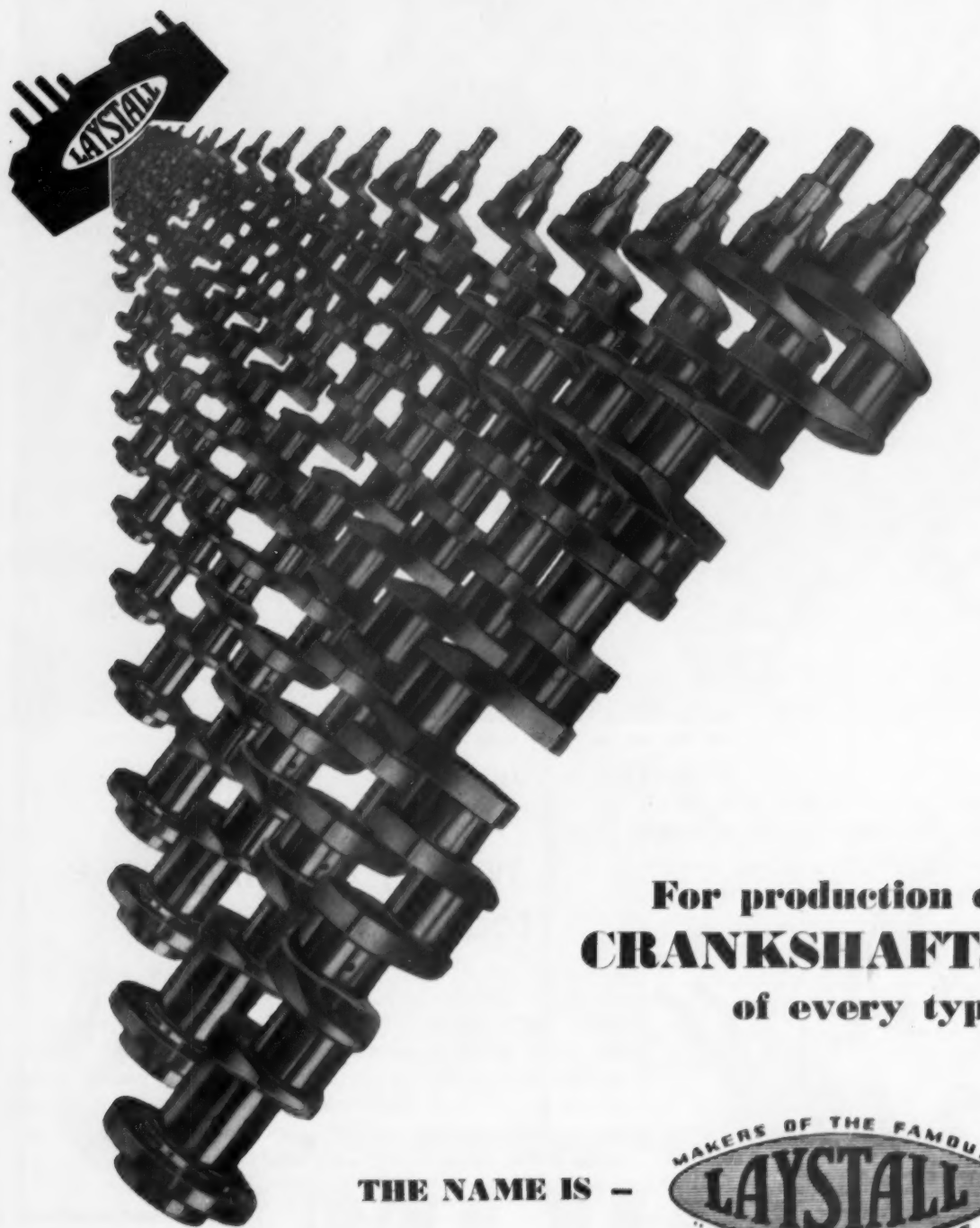
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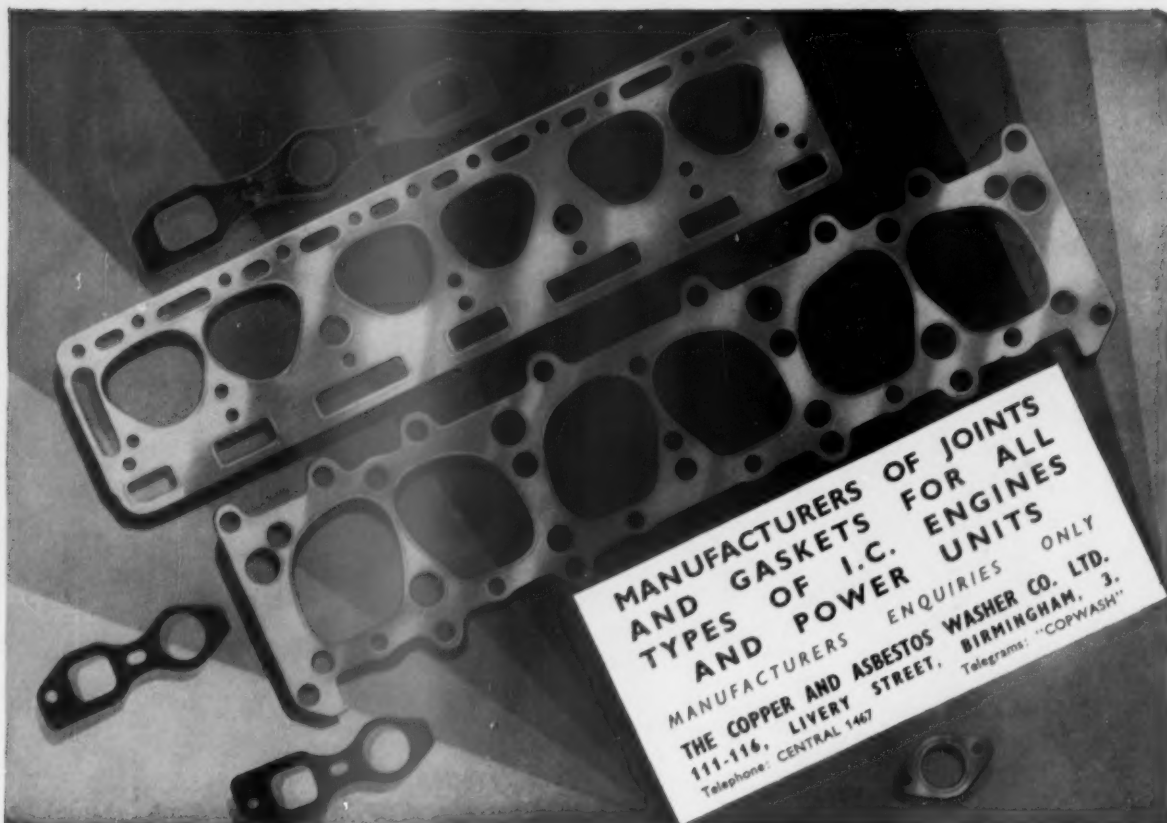
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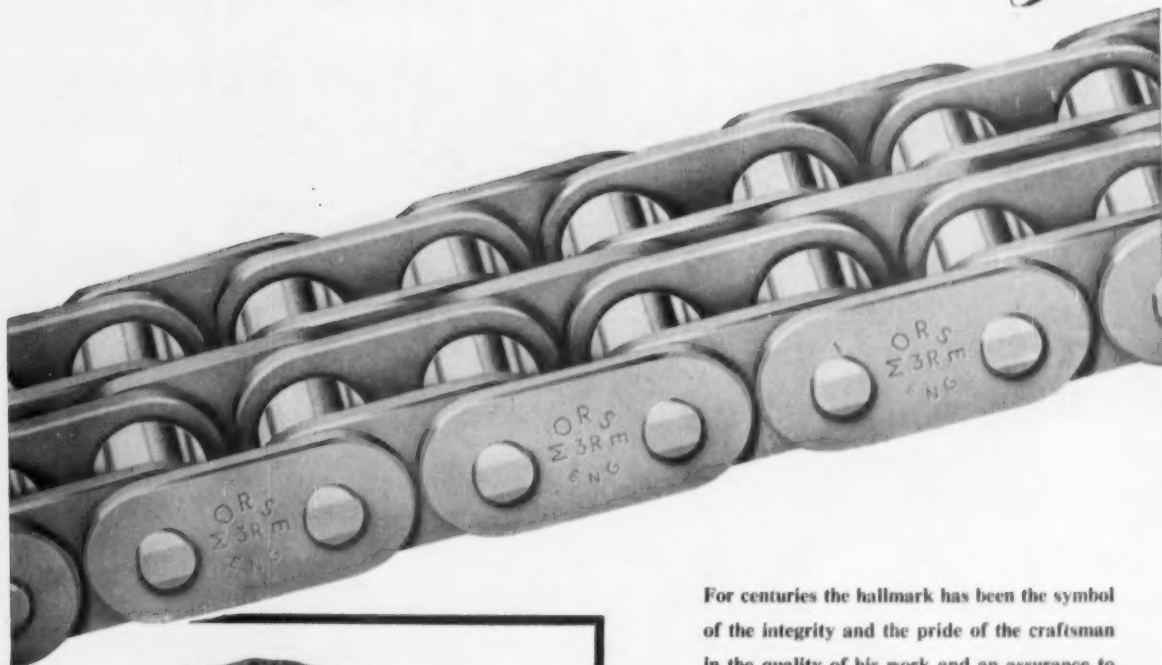
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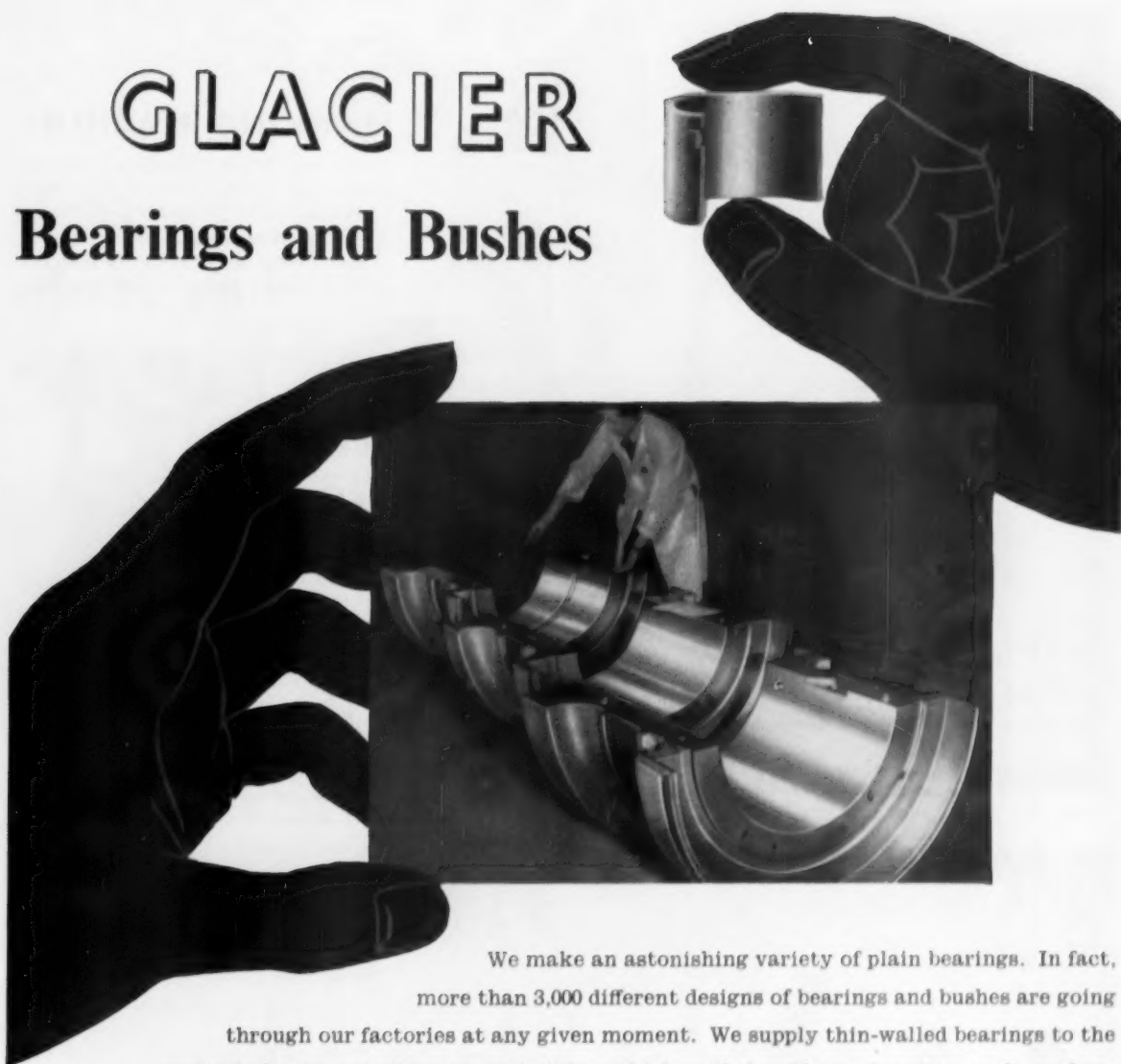
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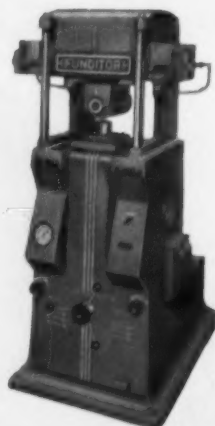
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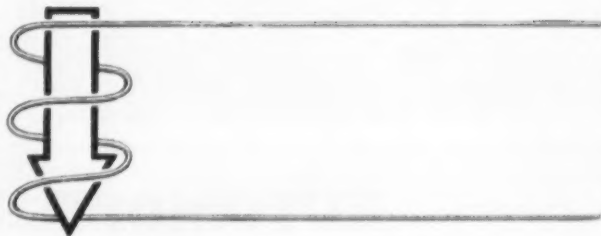
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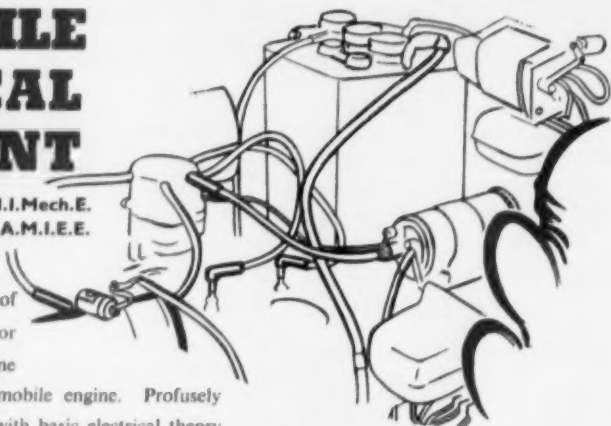
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TIS THE ROUNDHEADS, Sir Percy. Haste thee, Milord — jump for it!"

"But 'tis dark outside," said Sir Percy plaintively — off the battlefield he was a man of cautious disposition — "'tis wondering where to jump I am?"

"'Tis no time to ponder" retorted the Squire, "Haste!"

"Very well then, I suppose thou knowest best." Sir Percy grasped his sword, swung his cloak, and leapt into a world of darkness; he was pleasantly surprised to land on something soft and yielding. "Gad!" he beamed, "the age of miracles is not yet past." From beneath him, oddly muffled, the Squire's voice choked; "'Twould be uncommon kind of milord to remove his scabbard from the small of my back."

"Faith!" cried the cavalier — "here is a waggish situation indeed. Meseems I made a splendid landing on thy broad and faithful back. And where didst thou come to earth, old friend?"

The Squire squirmed uneasily. "Verily I could not swear to it, but if I mistake me not, we are in the cattleyard." His movements were accompanied by an odd squelching sound. "Verily thou wert unfortunate,"

said Sir Percy with soldierly sympathy — "perhaps that will teach thee the folly of a spring in the dark. And speaking of springs" — he spake with a knowledge surprising in one of that period — "Didst know that the most inferior springs may look exactly like a genuine high precision job, guaranteed for years of high performance? What! not interested in springs? Ah, well, perhaps a tub would suit thee better." He ambled off into the darkness . . .

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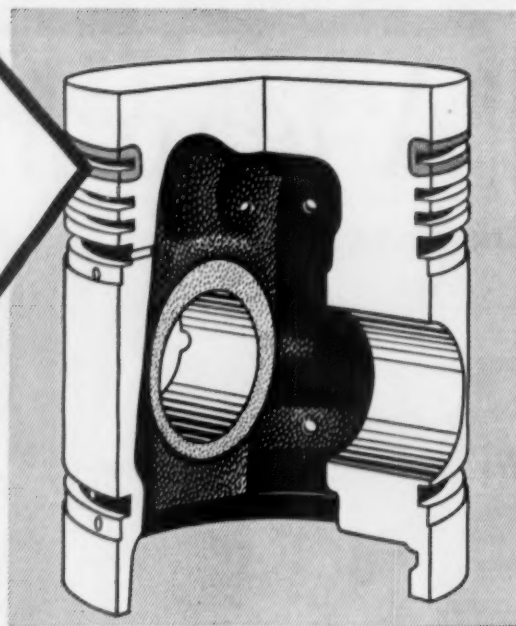
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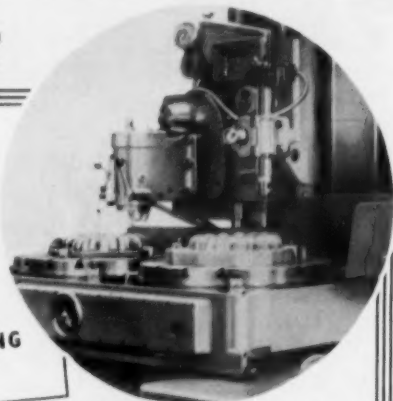


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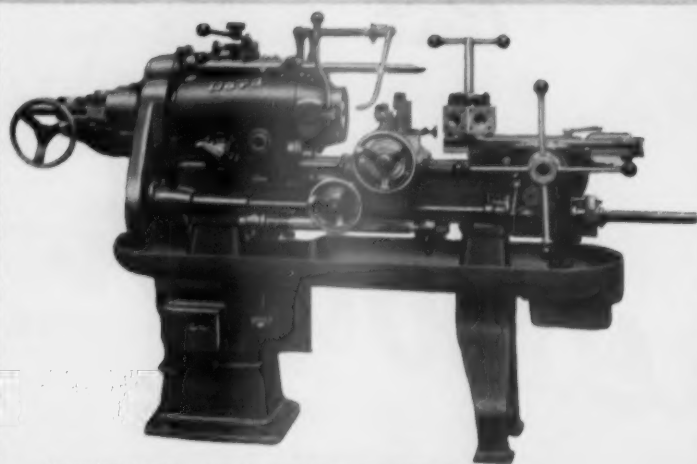
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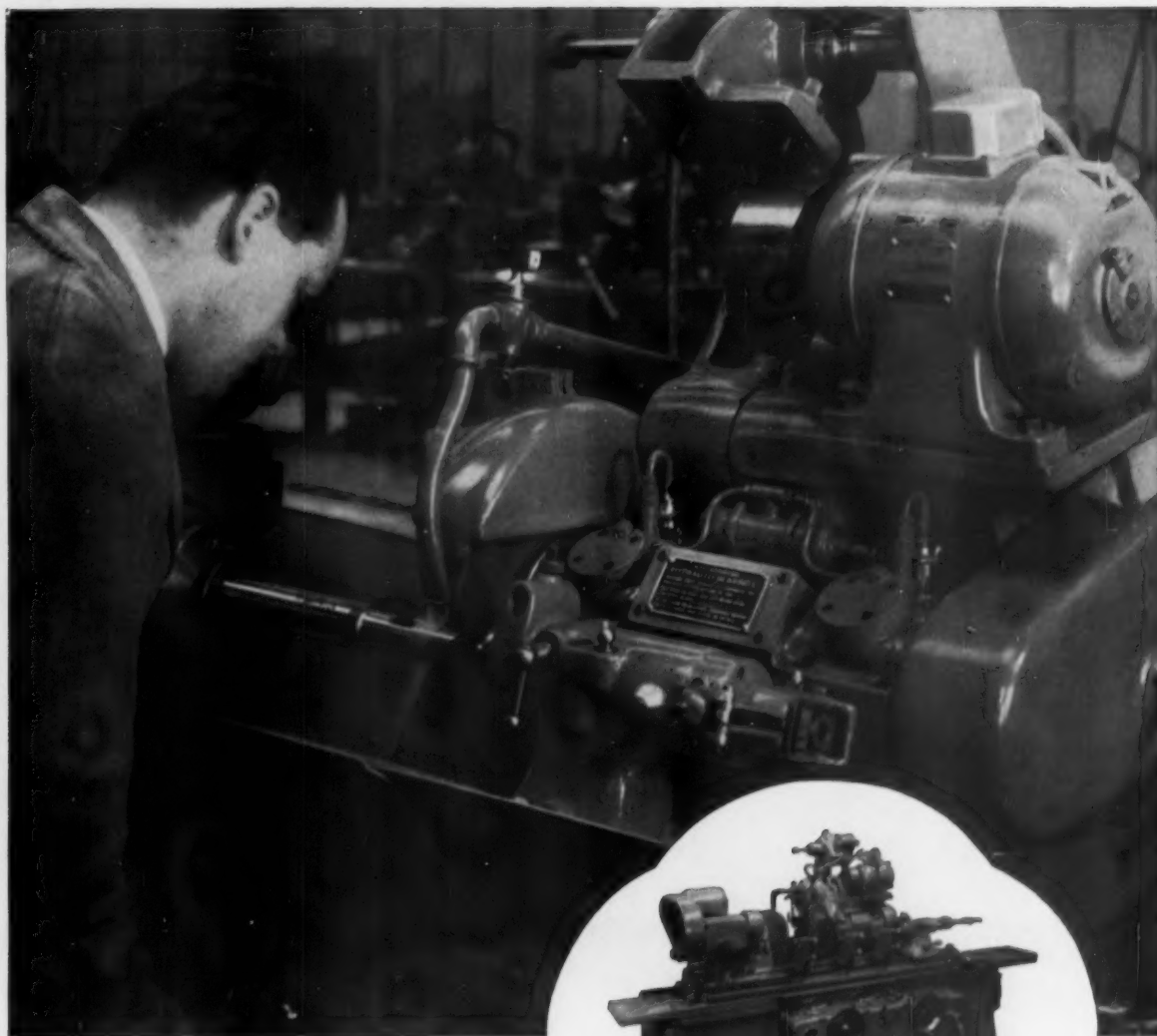
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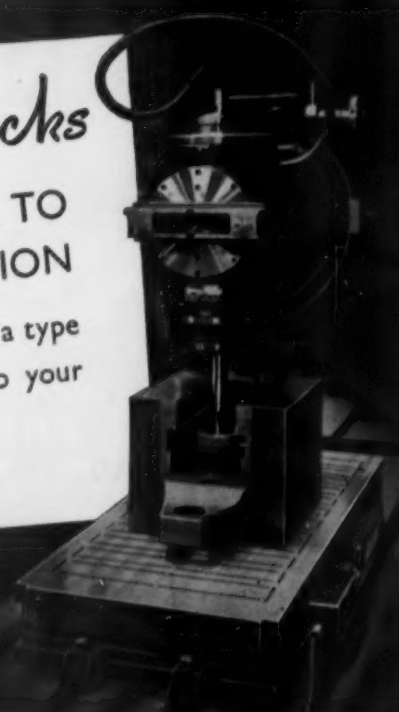
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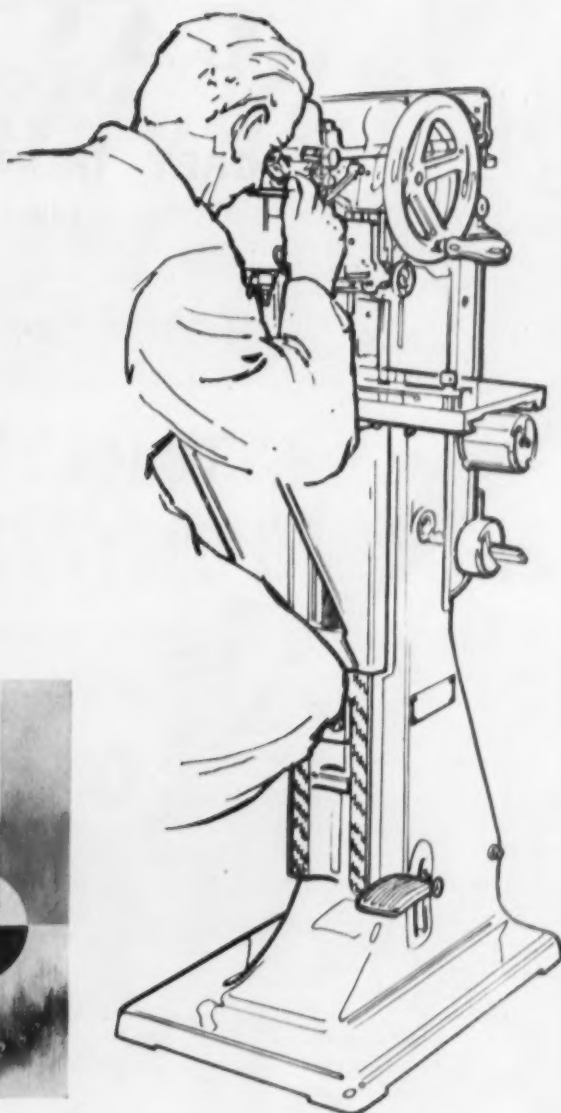
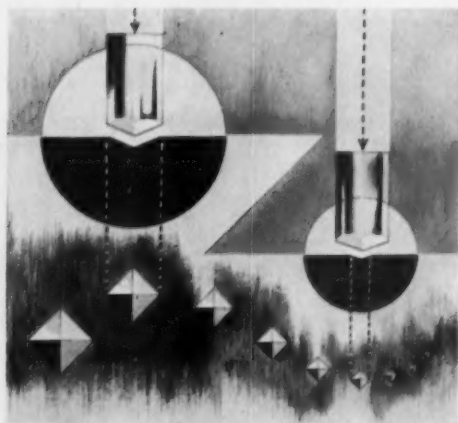
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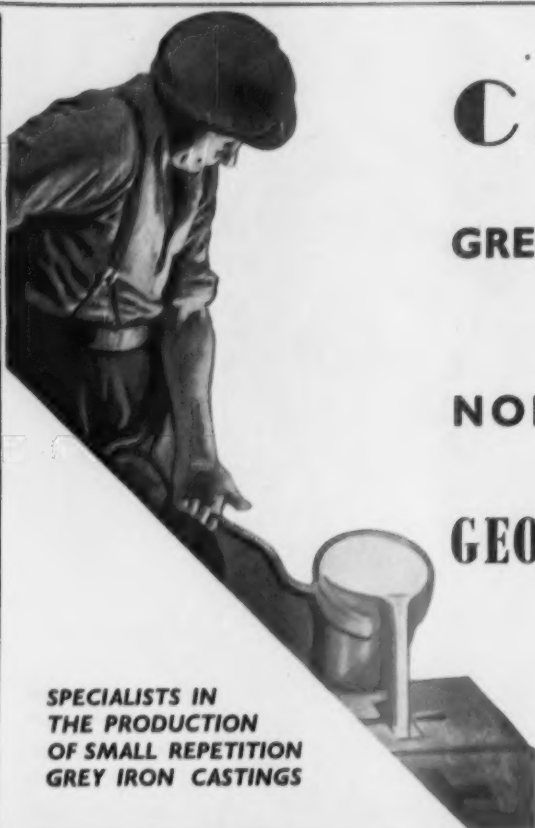
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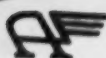
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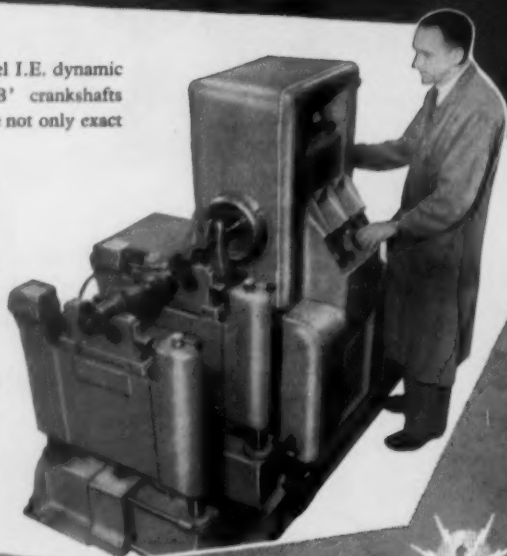
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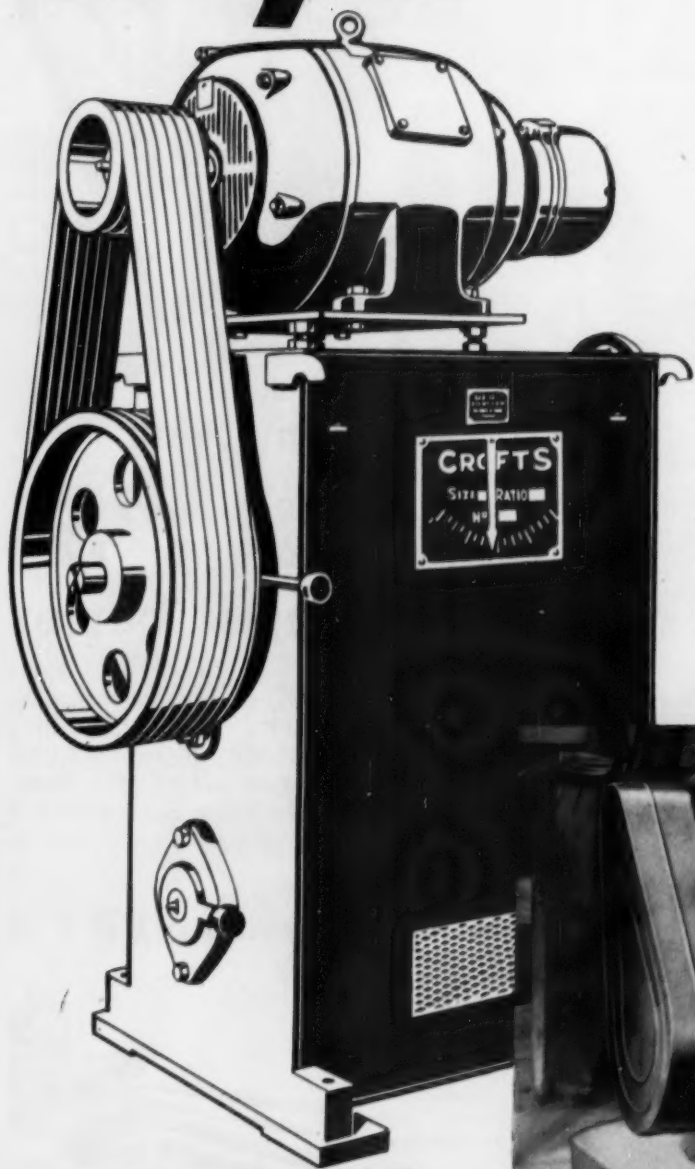
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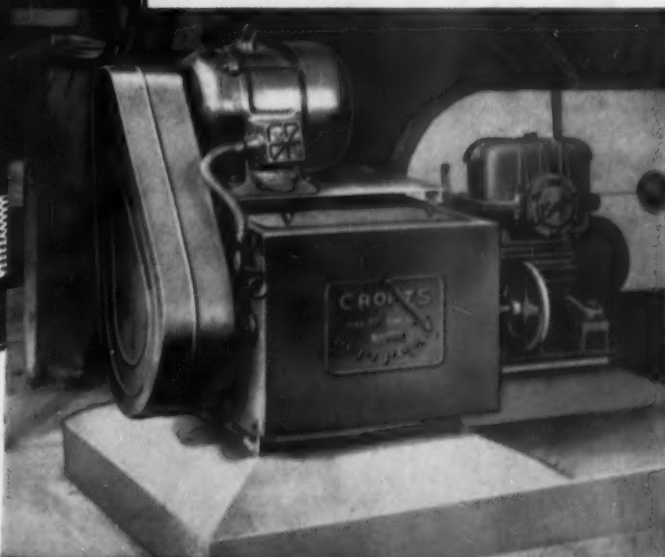
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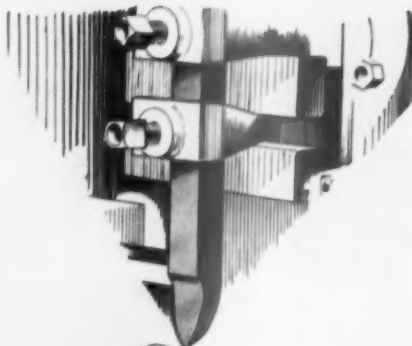
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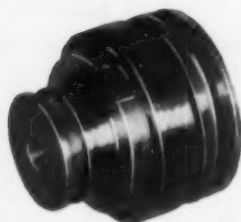
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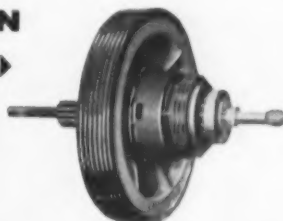


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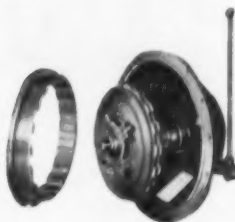
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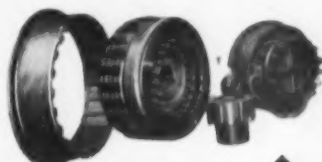
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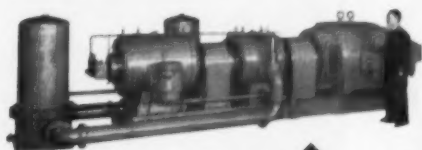
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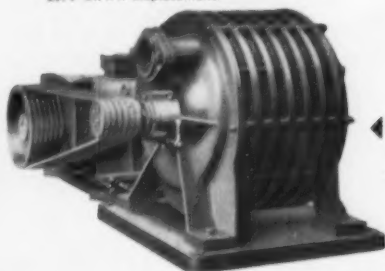
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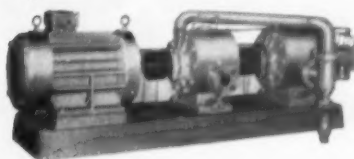


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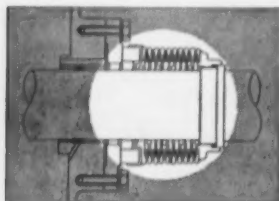
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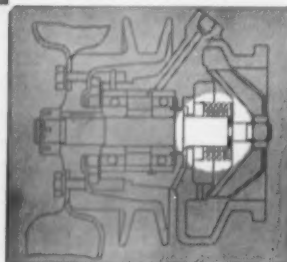
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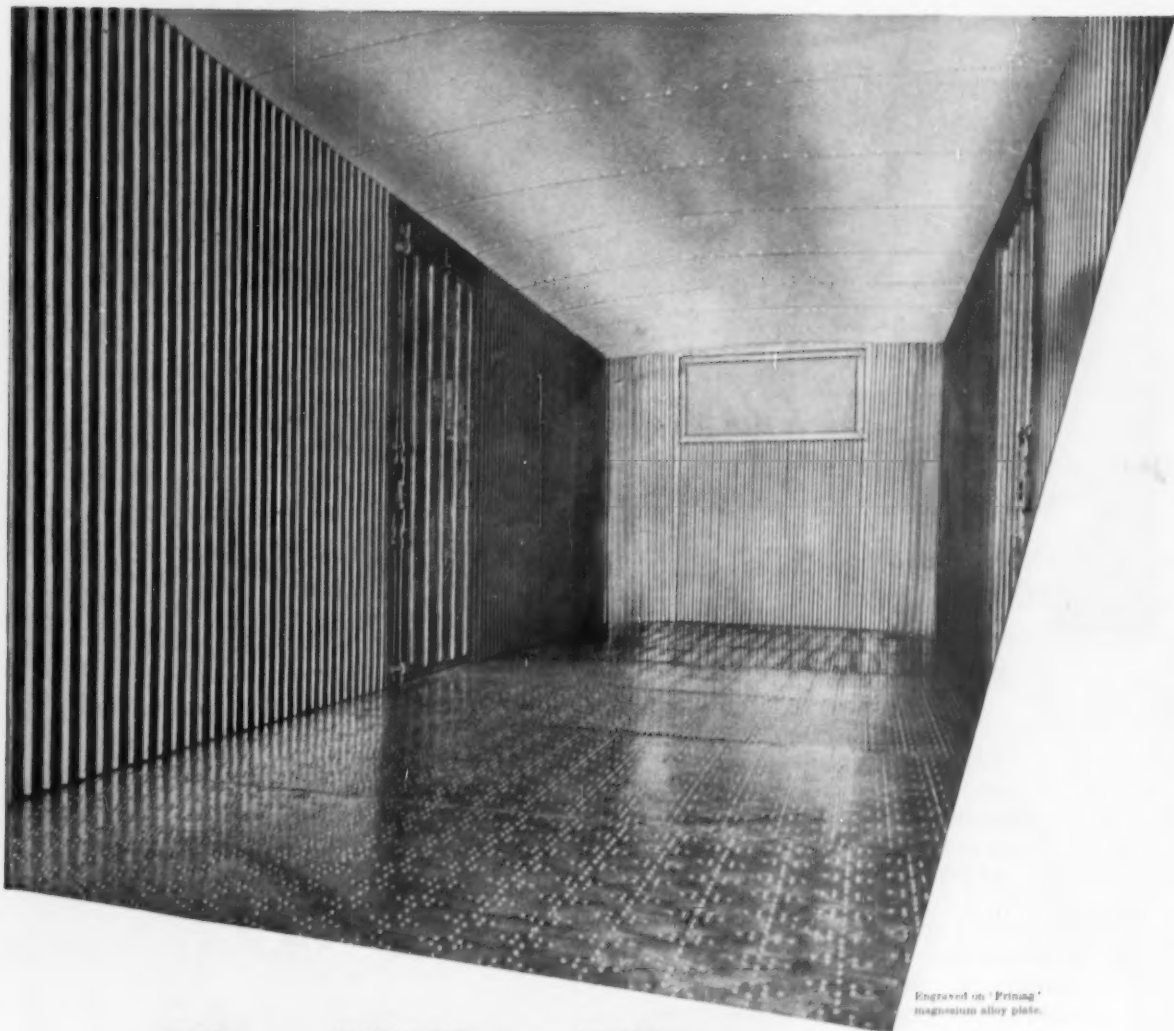


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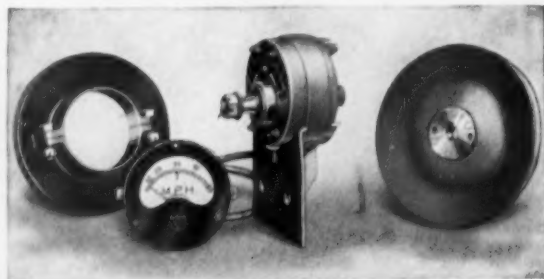
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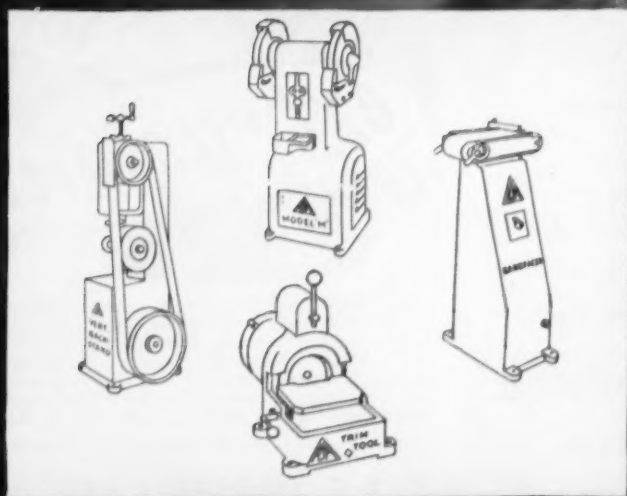
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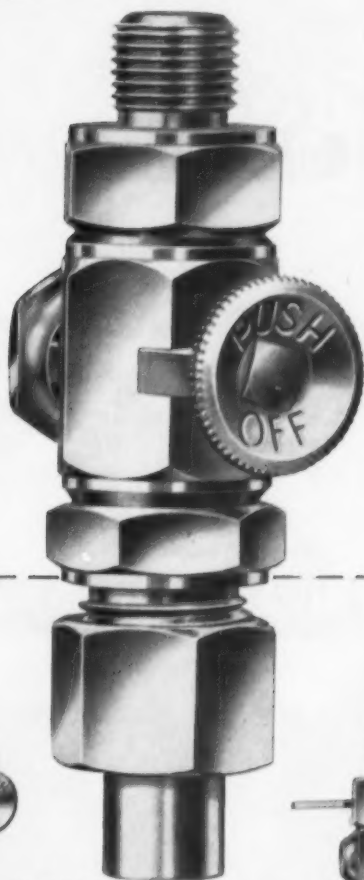


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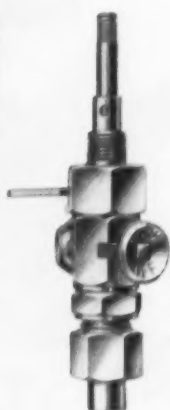
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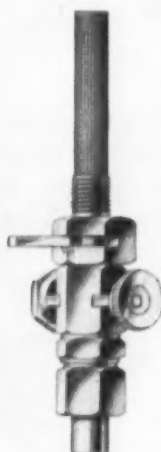
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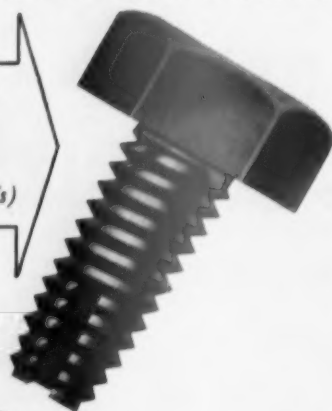
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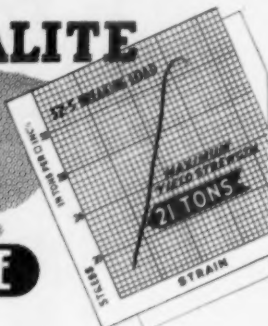
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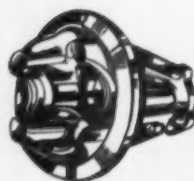
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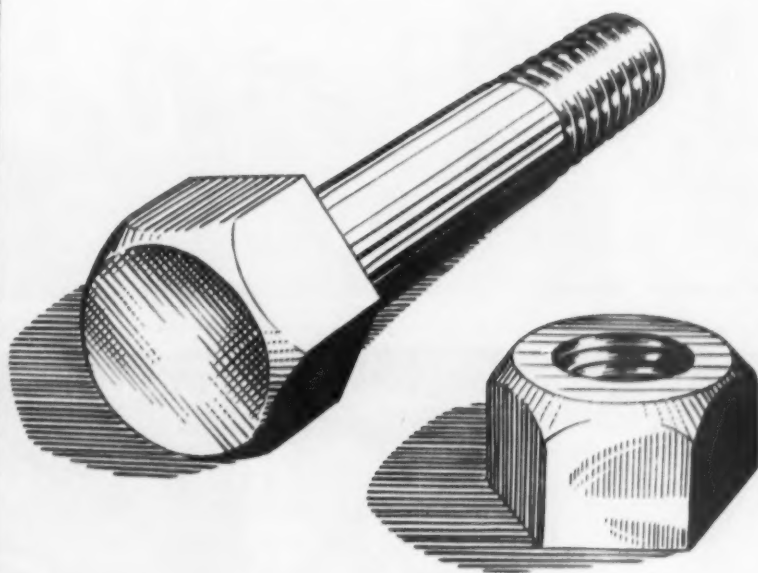
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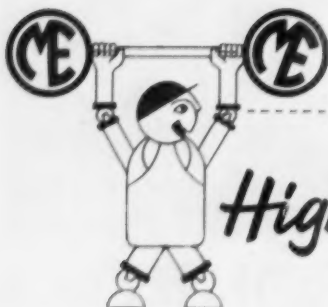
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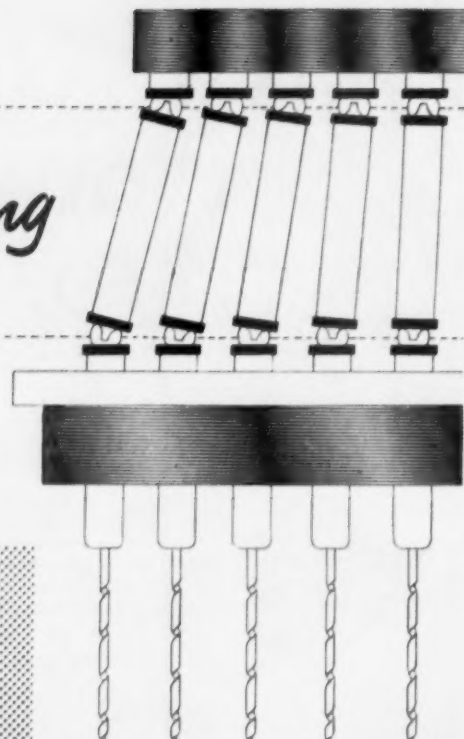
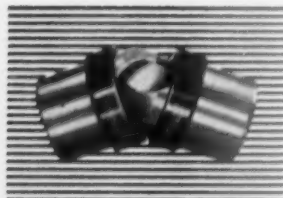
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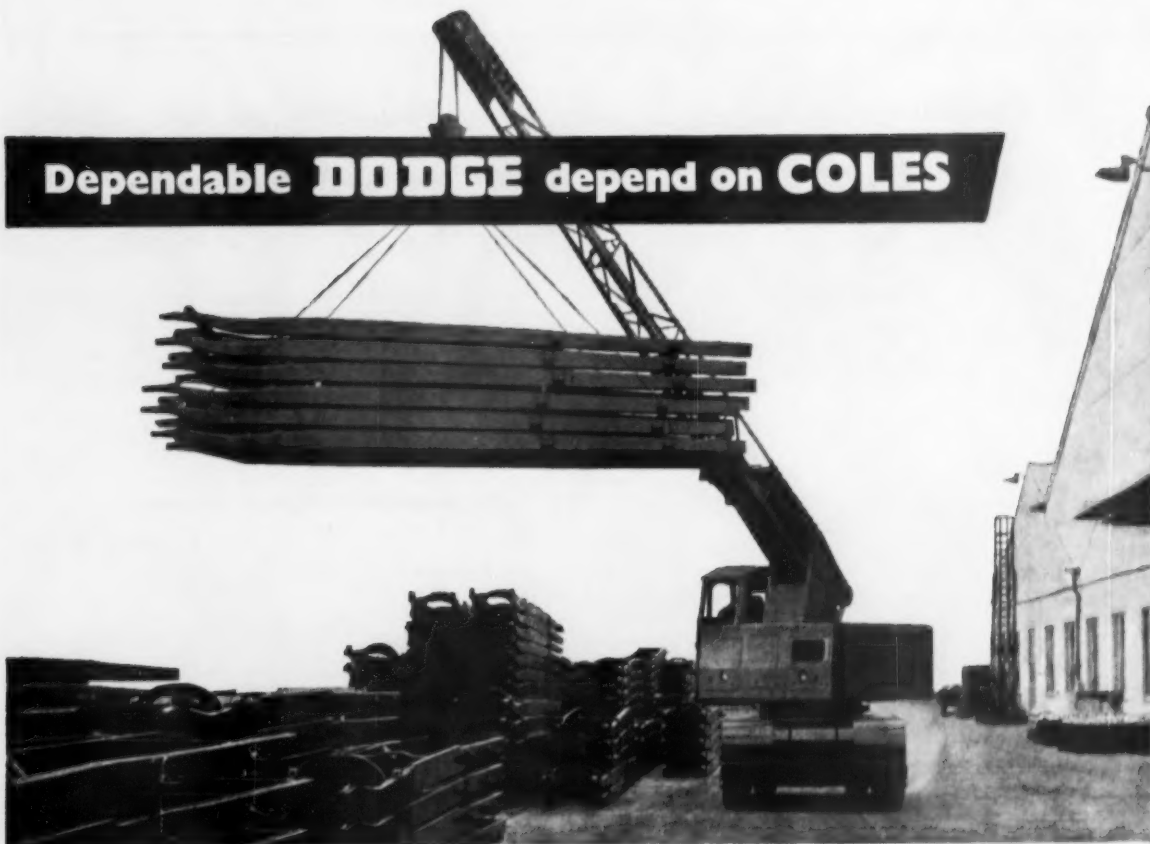
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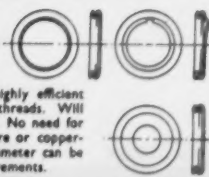
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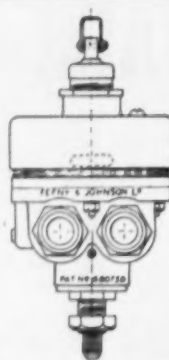
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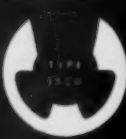
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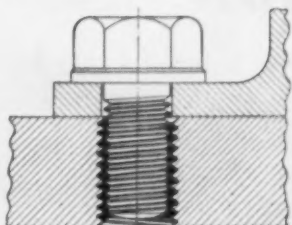
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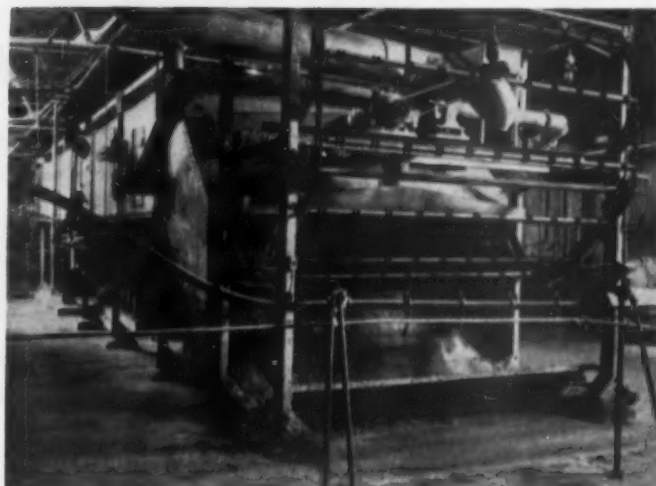
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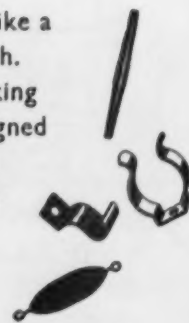
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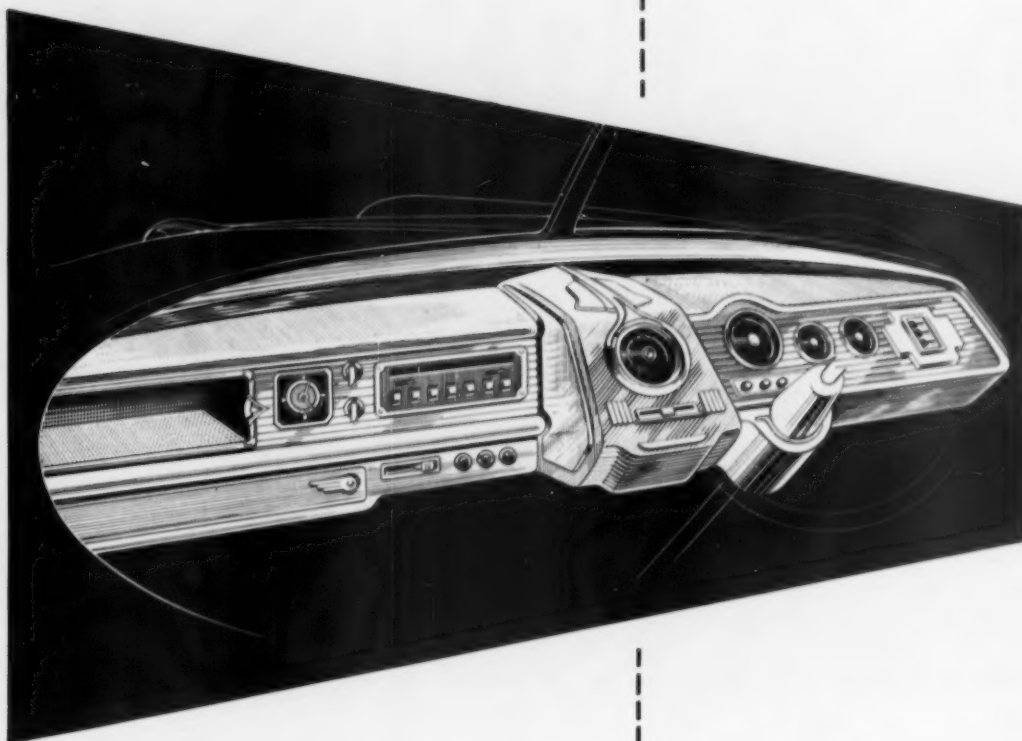
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Cutting a Dash . . .



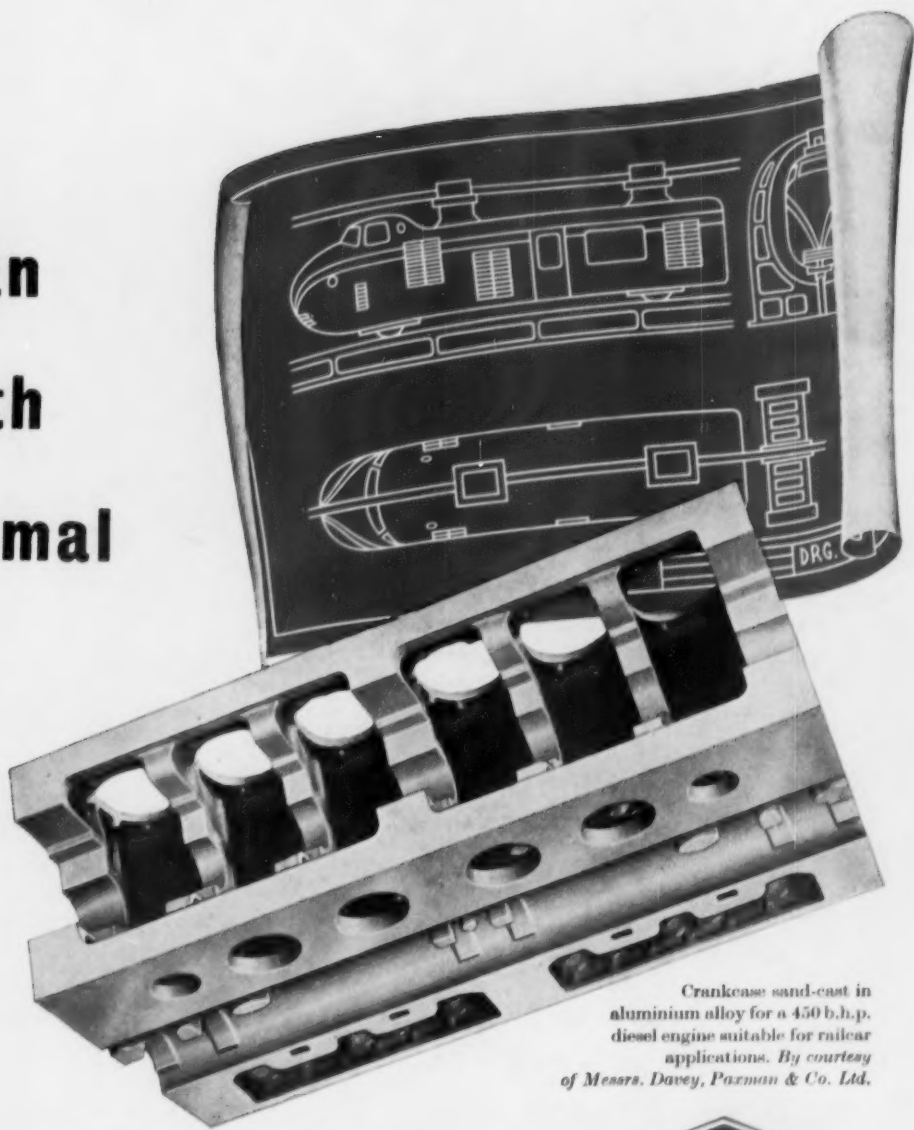
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